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Assessment of Emission Control Technology and Cost for Engines used in Handheld Equipment

CALIFORNIA ENVIRONMENTAL PROTECTION AGENCY

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Research Division

ASSESSMENT OF EMISSION CONTROL TECHNOLOGY AND COST FOR ENGINES USED IN HANDHELD EQUIPMENT

Final Report 93-324

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ABSTRACT

The California Air Resources Board (ARB) has established two tiers of emission standards for handheld utility engines. The Tier I standards took effect in 1995, and have been met with only simple engine and carburetor modifications. The more stringent Tier II emission standards will take effect in 1999, and are considered technology-forcing. Controversy has arisen over their technological feasibility, costs, and cost-effectiveness of the Tier II standards. To help resolve this controversy, Engine, Fuel, and Emissions Engineering, Inc. (EF&EE) was contracted by the Air Resources Board to examine two-stroke engine control technologies, the associated costs to manufacturers and consumers, and the cost-effectiveness of the resulting emission reductions. Building on earlier studies of two-stroke engines in motorcycles, EF&EE reviewed the technical literature on emissions and control technology for two-stroke engines and identified four potential technological approaches to meeting the Tier II standards, of which three are considered to have very high probability of success. These approaches are:

1. Direct in-cylinder fuel injection combined with a catalytic converter;

2. Port or crankcase fuel injection combined with a catalytic converter; and

3. Conversion to four-stroke, overhead-valve technology, with or possibly without a catalytic converter.

Less certain of success, but potentially much lower in manufacturing costs, is the fourth option, the use of stratified scavenging in conjunction with a catalytic converter. The use of a catalytic converter alone would not be feasible, since pollutant concentrations in present Tier I engines are high enough to overheat and destroy a catalytic converter. It is therefore necessary to reduce these concentrations substantially by other means before the catalytic converter can be durable and effective.

EF&EE estimated the increase in the retail price equivalent (RPE) of a typical item of handheld equipment using each of these approaches. The estimated RPE increase due to implementing these technologies ranges from $39 to $66 per unit. This increase would be offset by fuel cost savings of at least 30% (representing that portion of the fuel that is presently wasted out the exhaust in the form of HC emissions). For commercial handheld equipment, the fuel saving would be about $151, and for residential equipment about $6. EF&EE also calculated the cost-effectiveness of the Tier II standards, compared to continuing the present Tier I standards. For commercial equipment, the costs were negative, due to the fuel cost saving. Thus the Tier II regulations represent a "win-win" solution for commercial equipment. For residential equipment, if all lifecycle costs were allocated to the HC reduction, the costs per ton of HC eliminated would be between $4,129 and $7,531 per ton, depending on which technology is employed. This is within the range of cost-effectiveness for other VOC control measures that have been adopted by ARB.

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1. SUMMARY

As emissions from motor vehicles and stationary source of air pollution have been brought under control, pollution from non-road mobile sources such as utility equipment has been recognized as significant. Two-stroke gasoline engines used in handheld utility equipment are estimated to be responsible for 62% of all HC emissions from handheld and non-handheld utility equipment, as well as 55% of the PM emissions (ARB, 1990). Commercial users such as lawn and garden services and institutional grounds maintenance departments account for an estimated 85% of total emissions from handheld equipment, but only a small fraction of handheld equipment sales. Residential use by individual homeowners accounts for the rest.

The most important pollutant emissions from two-stroke engines are unburned hydrocarbons (HC), carbon monoxide (CO), and particulate matter (PM). Of these, HC is the most significant from an air-quality perspective. Compared to other engine types, two-stroke gasoline engines have extremely high HC emissions. These high emissions are partly due to the use of fuel-air mixture for scavenging, which allows up to 45% of total fuel used to pass unburned into the exhaust. Another important source of HC emissions is poor scavenging at idle and light loads, allowing too much exhaust to be retained in the cylinder, leading to poor combustion and misfire. CO emissions from two-stroke utility engines are due to the use of very rich air-fuel mixture settings on the carburetor. These rich mixtures are used to increase power output, to reduce operating temperatures in the cylinder, and to make the engine easier to start.

In order to bring utility equipment emissions under control, ARB established two tiers of emission standards for utility engines. The Tier I standards, which took effect in 1995, are fairly lenient. Many engines have been certified to meet the Tier I standards with only simple engine and carburetor modifications. The more stringent Tier II emission standards will take effect in 1999. To meet these standards is anticipated to require some form of exhaust treatment devices and/or advanced engine technologies.

Because of the technology-forcing nature of the Tier II standards, controversy has arisen over their technological feasibility and the costs and cost-effectiveness of compliance. To help resolve this controversy, Engine, Fuel, and Emissions Engineering, Inc. (EF&EE) was contracted by the Air Resources Board to examine two-stroke engine control technologies, the associated costs to manufacturers and consumers, and the cost-effectiveness of the resulting emission reductions. Building on earlier studies of two-stroke engines in motorcycles, EF&EE reviewed the technical literature on emissions and control technology for two-stroke engines and identified four potential technological approaches to meeting the Tier II standards, of which three are considered to have very high probability of success. These approaches are:

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Less certain of success, but potentially much lower in manufacturing costs, is the fourth option, the use of stratified scavenging in conjunction with a catalytic converter. It should be noted that the use of a catalytic converter alone would not be feasible, since pollutant concentrations in present Tier I engines are high enough to overheat and destroy a catalytic converter. It is therefore necessary to reduce these concentrations substantially by other means before the catalytic converter can be durable and effective.

EF&EE estimated the increase in the retail price equivalent (RPE) of a typical item of handheld equipment using each of these approaches. The RPE estimation followed the methodology developed by EPA. The estimated RPE increase due to implementing these technologies ranges from $39 to $66 per unit. Since any of these technologies would also reduce fuel consumption by at least 30% (representing that portion of the fuel that is presently wasted out the exhaust in the form of HC emissions), the life-cycle costs of these technologies to the consumer would be less than the initial RPE. For commercial handheld equipment, the fuel saving would be about $151, or several times the incremental cost. For residential equipment, the fuel saving would be about $6.

EF&EE calculated the cost-effectiveness of the Tier II standards, compared to continuing the present Tier I standards. For commercial equipment, the costs were negative, as the fuel saving would outweigh the higher initial cost. Thus, for commercial equipment, the Tier II regulations represent a "win-win" solution. For residential equipment, if all lifecycle costs were allocated to the HC reduction, the costs per ton of HC eliminated would be between $4,129 and $7,531 per ton, depending on which technology is employed. This is within the range of cost-effectiveness for other VOC control measures that have been adopted by ARB. Further, these costs per ton are probably overestimated, since they assume that all present buyers of engine-driven handheld equipment would continue to purchase engine-driven equipment at the higher costs. Faced with higher costs, many consumers would probably choose to purchase electric-powered handheld equipment instead. This equipment has zero direct emissions, and is also less costly than engine-driven equipment even at present prices.

2. INTRODUCTION

With the increasingly successful control of motor vehicle and stationary source emissions, off-road mobile sources such as utility engines have been recognized as important contributors to air pollution. A study carried out for the Air Resources Board (ARB) determined that handheld two-stroke utility equipment is responsible for 62% of total HC emissions from utility equipment, and for 55% of total PM emissions (ARB, 1990). Approximately 85% of the emissions from handheld utility equipment are produced by equipment in commercial, rather than home use.

In order to bring utility equipment emissions under control, ARB has established emission standards for utility engines. The Tier I standards, which were implemented in 1995, are fairly lenient. Many engines have been certified to meet the Tier I standards with only simple engine and carburetor modifications. The more stringent Tier II emission standards will take effect in 1999, and are anticipated to require some form of exhaust treatment devices and/or advanced engine technologies.

Since the adoption of these standards, continuing manufacturer concerns regarding the feasibility of the Tier II emission standards for handheld utility engines have led ARB to reassess the feasibility and cost-effectiveness of the Tier II standards. The present study forms a part of that effort. Engine, Fuel, and Emissions Engineering, Inc. (EF&EE) was contracted by ARB to review emission control technologies and costs to meet ARB's 1999 or Tier II emission standards for engines that used in handheld equipment. The objectives of the project were to:

1) review and compile engine performance and emission characteristics of existing (1995) handheld equipment, and review the latest technical literature on emission reduction techniques for small engines;

2) evaluate the technical feasibility of more traditional emission reduction techniques, such as improved fuel delivery systems, port revisions, and combustion management, for small two-stroke engines to meet the Tier II utility engine standards;

3) evaluate the technical feasibility of advanced emission reduction techniques, such as the use of catalytic converters and/or fuel injection systems, for small two-stroke engines to meet the Tier II utility engine standards;

4) evaluate the technical feasibility of utilizing small four-stroke engines for some handheld equipment-specifically, evaluate the design features and drawbacks of the Ryobi handheld, overhead valve four-stroke engines recently introduced in string trimmers;

5) estimate the incremental costs to the manufacturers and consumers of meeting the Tier II standards in each type of equipment.

This report documents the findings of the project. Following the introduction, Chapter Three presents the emission standards and test procedures for handheld equipment, as well as discussing issues related to the test cycles. Chapter Four describes the technology and emission characteristics of small two-stroke and four-stroke engines. It also presents emission levels from uncontrolled handheld equipment and for handheld equipment certified to the Tier I (1995) emission standards) Smoke and particulate emissions from these engines are also discussed in this chapter. A comprehensive review of emission control technologies for small two-stroke engines is presented in Chapter Five. Chapter Six presents several design alternatives for an engine to meet the ARB Tier II standards, and assesses the incremental costs to consumers associated with each alternative.

3. EMISSION STANDARDS AND TEST PROCEDURES FOR HANDHELD EQUIPMENT

3.1 Utility Engine Emission Standards

The emission regulations for utility, and lawn and garden engines were approved by the ARB in 1990. The Tier I regulations were originally applicable to engines produced on or after January 1, 1994. However, a one year delay in implementation was approved by the ARB on April 1993, so that the regulations came into force for new engines produced on or after January 1, 1995. The Tier II standard is set to be implemented beginning January 1, 1999. Table 1 shows both the Tier I and the Tier II regulations adopted by ARB. As Table 1 shows, the emission standards are different for handheld and non-handheld equipment. Some common handheld and non handheld equipment are listed in Table 2.

Experience since the adoption of the ARB regulations has shown that the Tier I standards can be met fairly readily through simple engine modifications such as fuel-air ratio changes, calibrations, and reductions in component tolerances. Certification emission data from a number of small handheld equipment are presented later in this report. The Tier II standards for handheld equipment, however, are considered to be very challenging. According to the industry, the PM standard is the most difficult one to meet.

Table 1: CARB emission standards for utility engines.

Non-Handheld Equipment			
Emissions (g/BHP-hr)	THC + NOx	CO	Diesel PM
1995 (Tier I)			
< 225 cc	12.0	300	0.9
≥ 225 cc	10.0	300	0.9
1999 on (Tier II)	3.2	100	0.25

Handheld Equipment				
Emissions (g/BHP-hr)	THC	NOx	CO	PM
1994-1998 (Tier I)				
< 20 cc	220	4.0	600	-
20 - 50 cc	180	4.0	600	-
≥ 50 cc	120	4.0	300	-
1999 on (Tier II)	50	4.0	130	0.25

Table 2: Some common handheld and non handheld utility equipment.

Common Small Utility Equipment	
Handheld Equipment	Non Handheld Equipment
Chainsaws	Lawnmowers
Blowers	Lawn and Garden Tractors
Vacuums	Rear Engine Riders
Sprayers	Shredders
String Trimmers	Snow Blowers
Hedge Trimmers	Tillers
Brush Cutters	Edgers
Cut-Off Saws	Chippers
Engine-Driven Drills	Commercial Turf Equipment
Handheld Edgers	Front Mowers
	Wood Splitters
	Grinders
	Air Compressors
	Concrete Cutters
	Gas Compressors
	Generator Sets
	Pumps
	Welding Machines

3.2 Utility Engine Test Procedure

The small utility engine test procedures are based on the procedures given in Society of Automotive Engineers (SAE) Recommended Practice J1088. The SAE procedures were first published in 1974 by the SAE Small Engine Emissions Subcommittee, and later finalized in 1983, with the purpose of specifying uniform procedures for the evaluation of exhaust emissions from small spark-ignition utility engines. The test procedures comprise a set of steady-state operating modes with different load and speed conditions. These modes are intended to reflect the range of typical operating conditions for small utility equipment. The test cycle for a specific utility engine will consist of a set of set of modes that represent the typical operating conditions of that particular engine or equipment. The overall results of the emission test are given by the weighted sum of the results from each mode.

Three test cycles, each consisting of a specified set of the SAE J1088 modes, have been adopted by ARB. These test cycles are presented in Table 3. Test cycle A is for non-handheld equipment engines that are configured by the engine manufacturer to operate primarily at an intermediate speed. Test Cycle B is for non-handheld equipment engines that are configured by the engine manufacturer to operate primarily at rated speed. Test Cycle C is for handheld equipment engines.

As Table 3 shows, test cycle C contains only two modes: idle, and full load (wide open throttle at the maximum power speed identified by the engine manufacturer). The modal weighting factors are 90% for the full load mode and 10% for the idle mode. Emissions in each mode are measured in grams per hour, and power output is measured in brake horsepower (note that power output at idle is zero). These data are then combined according to the weighting factors to calculate cycle-composite emissions in grams per brake horsepower-hour (g/BHP-hr).

$$Emissions\ (g/BHP\text{-}hr) = \frac{0.9\ x\ grams/hr_{fullpower} + 0.1\ x\ grams/hr_{idle}}{0.9\ x\ HP_{fullpower} + 0.1\ x\ 0}$$

ARB test procedures allow emissions (in grams per hour) to be measured either by the raw gas method (RGM) or the constant volume sampling (CVS) method. The RGM test procedures are based on those specified in SAE J1088, while the CVS test procedures are based on those used for motor vehicle testing. Since the Tier II emission standards require measurement of PM

Table 3: ARB test cycles for small utility engines.

ARB Test Cycles for Small Utility Engines											
Mode	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**
Speed	**Rated Speed**					**Intermediate Speed**					**Idle**
Test Cycle A: Non-Handheld Equipment Engine, Primarily Operate at an Intermediate Speed											
Mode Point						1	2	3	4	5	6
Load (%)						100	75	50	25	10	0
Weighting (%)						9	20	29	30	7	5
Test Cycle B: Non-Handheld Equipment Engine, Primarily Operate at a Rated Speed											
Mode Point	1	2	3	4	5						6
Load (%)	100	75	50	25	10						0
Weighting (%)	9	20	29	30	7						5
Test Cycle C: Handheld Equipment Engine											
Mode Point	1										2
Load (%)	100										0
Weighting (%)	90										10

emissions, the CVS method is more convenient. Accurate measurement of particulate matter (PM) emissions requires that the exhaust be diluted to a temperature less than 53°C at the filter face, in order to assure that the heavy-hydrocarbon component of the PM has condensed. This can be accomplished easily in a CVS system, while the RGM requires a separate dilution tunnel for PM.

3.3 Test Cycle Issues

The ARB two-mode test procedure is not completely representative of handheld equipment operation. Most handheld equipment operates nearly all of the time either at idle or at wide-open throttle (WOT), but - except for blowers - wide-open throttle operation does not take place at a fixed speed. In addition, the transients between idle and WOT operation may be important for emissions.

For chainsaws, in particular, the duty cycle in actual operation differs considerably from ARB's two-mode test procedure. Chainsaws tend to spend much more than 10% of their time at idle in between cuts, and correspondingly less time at WOT. Thus, the 90:10 weighing does not reflect the actual balance between the modes. Furthermore, actual speed at WOT is not fixed at the maximum power speed, but varies considerably depending on the cutting power requirements. Before beginning a cut, the chain is normally accelerated at WOT with no load, and under these conditions the speed usually exceeds 12,000 RPM. When a cut is completed, the load on the engine drops, and the engine again accelerates rapidly to the WOT - no load speed. Finally, when the operator releases the trigger, the speed drops back to idle, but the inertia of the chain and clutch assembly is such that the engine is "motored" for a second or so. In debranching, the saw may cycle repeatedly from idle to high speed and back every few seconds. String trimmers may also experience significant transient operation.

Because of this transient operation, in-use emissions may differ significantly from the results of the two-mode, steady-state test. This raises the possibility that manufacturers may engage in "cycle beating" - designing their engines to have low emissions at the maximum power point, but not necessarily elsewhere in the operating range. Because of the dependence of two-stroke emissions and performance on gas-dynamic effects, which are strongly affected by engine speed, this problem is potentially more serious for a two-stroke engine than for typical four-strokes.

Researchers at the University of Michigan (UOM) found that emission levels from several handheld equipment engines tested on the two-mode test cycle (SAE J1088 C) with a 70:30 weighing, which the EPA and UOM's researchers referred as the "Chainsaw Cycle", were higher that those with a 90:10 weighing (Sun et al., 1995). Sun et al. also reported that the emission levels were higher if the engines were tested on transient cycles, such as the continuous SAE J1088C or Graz University of Technology (GUT) test cycles.

4. SMALL TWO-STROKE UTILITY ENGINES

Almost all engines used in small utility equipment are small, air-cooled, reciprocating Otto-cycle engines using gasoline fuel. These small engines can be separated into two-stroke and four-stroke designs. The distinction between two-stroke and four-stroke small engines is an important one for emissions, as two-stroke engines tend to emit much greater amounts of unburned hydrocarbons (HC) and particulate matter (PM) than four-stroke engines of similar size and power. Two-stroke engines also display markedly poorer fuel economy than four-strokes, but tend to have higher power output, quicker acceleration, and lower manufacturing costs. Because of their advantages in performance and manufacturing cost, two-stroke engines are used extensively in small utility equipment where this is permitted by emission regulations.

Except for battery-electric or cord-electric equipment, nearly all hand-held utility equipment such as chainsaws, trimmers and blowers is powered by crankcase-scavenged two-stroke engines. Reasons for using two-stroke engines include compactness and the ability to operate in a variety of positions, including upside down, as well as better power-to-weight ratio and lower manufacturing cost. These engines range from 20 to 100 cc displacement. Recently, a handheld engine and equipment manufacturer, Ryobi, has introduced string trimmers powered by a four-stroke engine with overhead valves. The Ryobi four-stroke engine is discussed later in Chapter Five.

The following sections discuss the general operating principles of two-stroke and four-stroke engines. Although the focus of the study was on controlling emissions from small two-stroke engines, the operating principles of four-stroke engines are also discussed. Switching to four-stroke engines is one alternative for meeting ARB's Tier II standards for hand-held utility equipment.

4.1 Operating Principles of Small Engines

Four-Stroke Engines

Figure 1 is a diagram of the piston and cylinder of a typical small overhead-valve four-stroke engine. Engine operation takes place in four distinct steps: intake, compression, power, and exhaust, with each step corresponding to one "stroke" of the piston, or 180° of crankshaft rotation. During the intake stroke, the intake valve opens to admit a mixture of air and fuel,

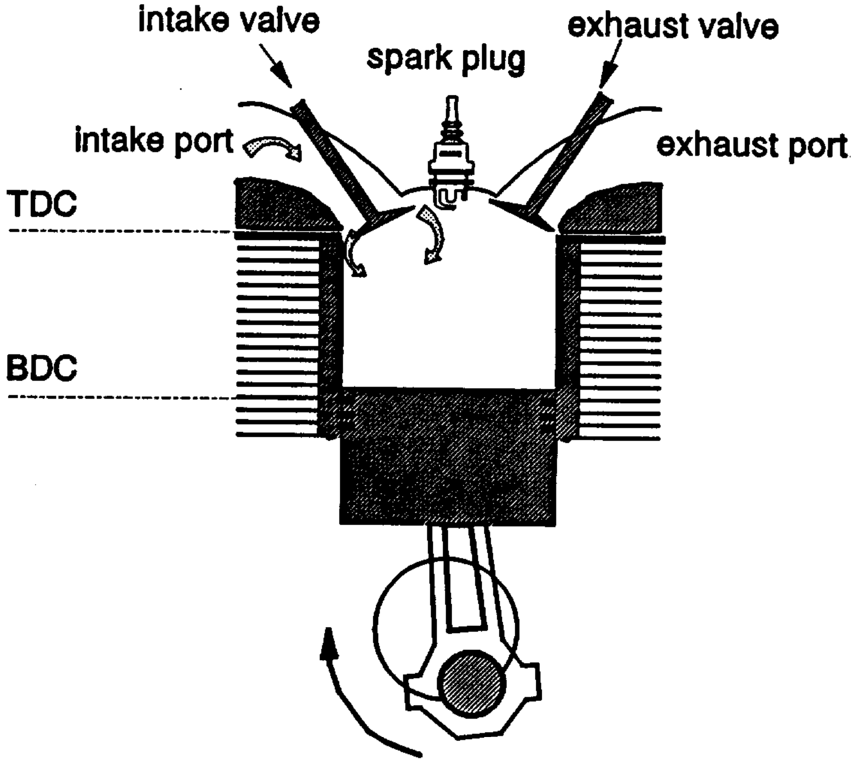


Figure 1: Diagram of an overhead valve, four-stroke engine.

which is drawn into the cylinder by the vacuum created by the downward motion of the piston. Figure 1 shows the piston near the end of the intake stroke, approaching bottom-dead-center. During the compression stroke, the intake valve closes, and the upward motion of the piston compresses the air-fuel mixture into the combustion chamber between the top of the piston and the cylinder head.

The compression stroke ends when the piston reaches top-dead-center. Shortly before this point, the air-fuel mixture is ignited by a spark from the spark plug, and begins to burn. Combustion of the air-fuel mixture takes place near top-dead-center, increasing the temperature and pressure of the trapped gases. During the power stroke, the pressure of the hot burned gases pushes the piston down, turning the crankshaft and producing the power output of the engine. As the piston approaches bottom-dead-center again, the exhaust valve opens, releasing the pent-up burned gases. Finally, during the exhaust stroke, the piston once more ascends toward top-dead-center, pushing the remaining burned gases in the cylinder out the open exhaust port as it does so. Near top-dead-center again, the exhaust valve closes and the intake valve opens for the next intake stroke. In a four-stroke engine, combustion and the resulting power stroke occur only once every two revolutions of the crankshaft.

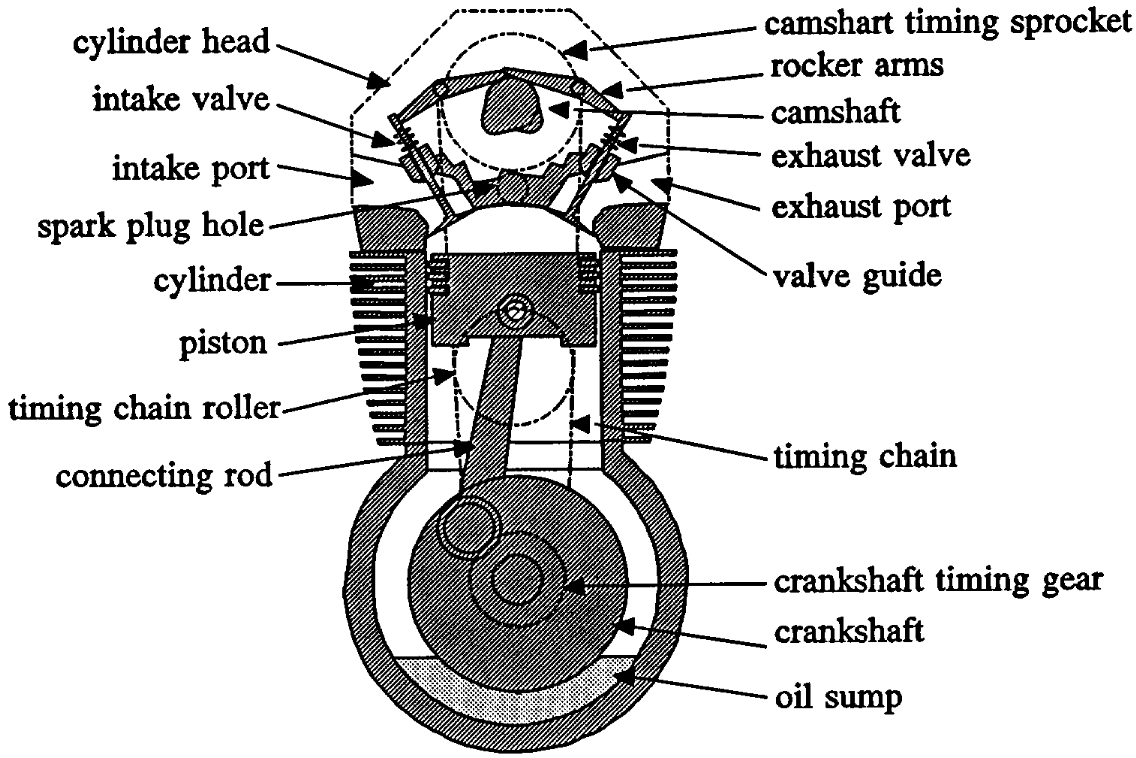


Figure 2: Mechanical layout of a typical overhead valve, four-stroke engine.

The mechanical systems required by four-stroke engines to open and close their intake and exhaust valves at the right time make these engines relatively complex to manufacture. Figure 2 shows the mechanical layout of a typical four-stroke engine. The valves are opened by lobes on the camshaft, which is driven at one-half engine speed by a sprocket and chain arrangement from the crankshaft. The camshaft lobes press on the valve followers, pushing up on the rocker arms, and causing the valves to open at the appropriate times in every second crankshaft revolution. The camshaft, valve linkage, crankshaft bearings, and pistons are lubricated by oil pumped from the oil sump at the bottom of the crankcase through a series of oil galleries. Since the camshaft is located above the cylinder, this is an "overhead cam" engine.

Another overhead valve design has the camshaft assembly located at the bottom of the cylinder near the crankshaft (see Figure 3). As shown in Figure 3, the camshaft gear is driven by a crankshaft gear, and two cam followers in the cam assembly translate the circular motion to linear motion. This linear motion is imparted to pushrods, which act on the rocker arms to open the valves. This design is easier to manufacture than an overhead cam engine, but does not perform as well as high RPM. The Ryobi handheld four-stroke engines uses this type of pushrod design.

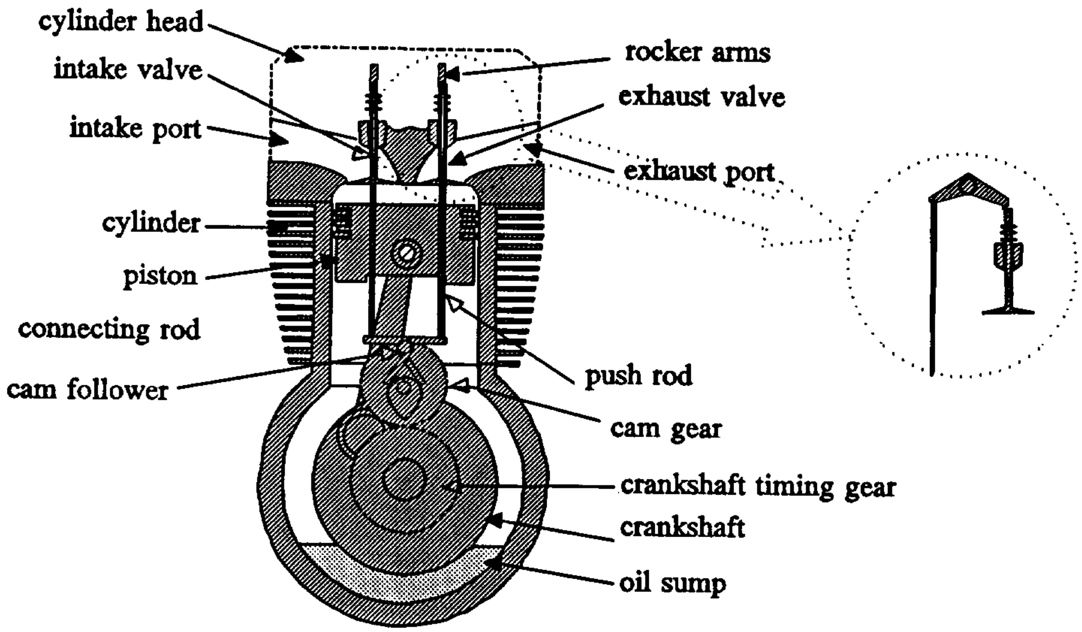


Figure 3: A typical overhead valve, four-stroke engine with push rod design.

Before the advent of emission regulations for utility engines, most four-stroke engines found in non-handheld utility equipment used a side-valve configuration. For side-valve engines, the intake and exhaust ports are located at the bottom of an extension of the combustion chamber, which projects out of the line of the cylinder bore. This makes it possible to eliminate the rocker arms, and drive the valves directly from the camshaft, thus reducing costs. The main drawback is the long, flattened combustion chamber that this requires. This results in greater heat transfer to the engine compared to an overhead valve engine, and a greater tendency to knock. Thus, the side-valve engine needs to run at a richer air/fuel mixture to prevent the engine from overheating, and at a lower compression ratio to prevent knock. The rich operation of the side-valve engine causes it to produce higher HC and CO emissions as compared to an overhead valve configuration. Also, the combustion chamber design of the side-valve engine provides higher surface area and crevice volumes for the fuel to settle in. Due to the fact that the flame is unable to penetrate, the fuel settled into these crevices becomes unburned HC emissions in the exhaust. Emissions data comparing overhead-valve and side-valve engines show that HC emissions from overhead-valve engines are about 30-50% lower. Also, since side valve engines have slow combustion, thus they are not suitable for engines operating at high RPM, such as those found in handheld equipment.

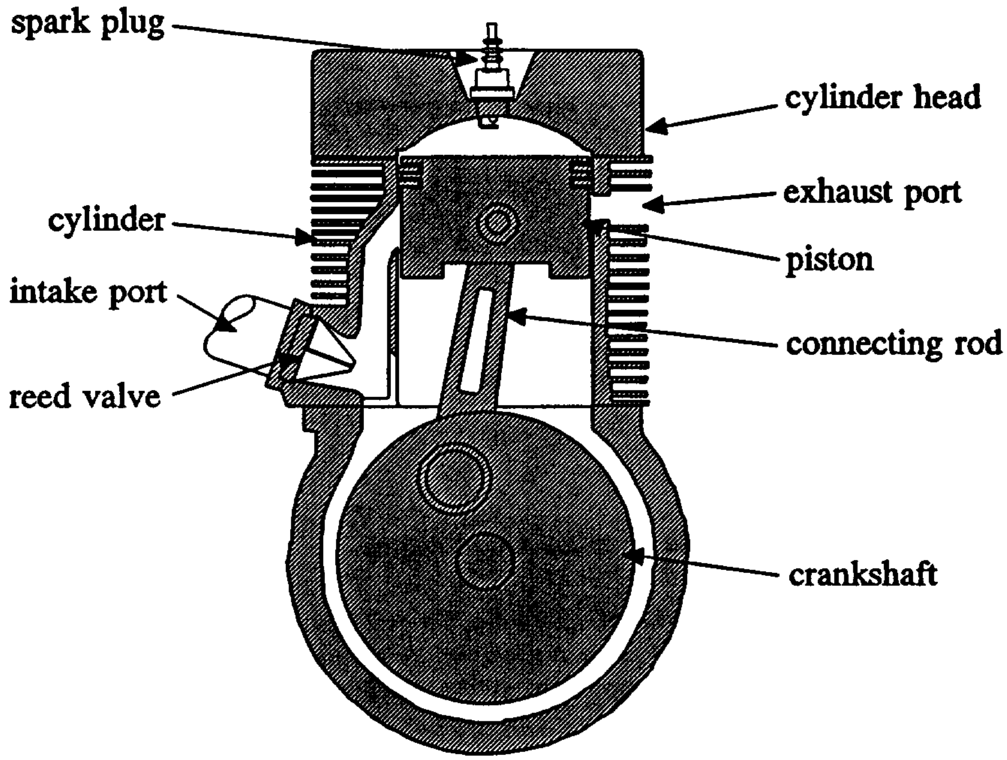


Figure 4: Mechanical layout of a typical two-stroke engine.

With the effectiveness of ARB's Tier I utility equipment regulations in 1995, many utility engines have been redesigned to use an overhead valve configuration. More than 100 overhead valve engine models were certified with ARB in 1995. One of them was the Ryobi 4-stroke handheld engine used in trimmers.

Two-Stroke Engines

A two-stroke small engine can be much simpler mechanically, as Figure 4 shows. The operation principal is very simple as well. Blair (1990) provides an excellent and very thorough discussion of two-stroke engine design and operation. Four stages in the combustion cycle of a simplified two-stroke engine are shown in Figure 5. In the first stage (Figure 5a), near the top of the compression stroke, the compressed charge in the cylinder is about to be ignited by the spark plug. At the same time the partial vacuum created by the rising piston draws fresh air-fuel mixture into the crankcase. Ignition is followed by combustion, and the pressure of the hot burned gases forces the piston down. As the piston approaches the bottom of the cylinder, the exhaust port in the wall of the cylinder is uncovered, and the combustion gases "blow down" into the exhaust port (Figure 5b).

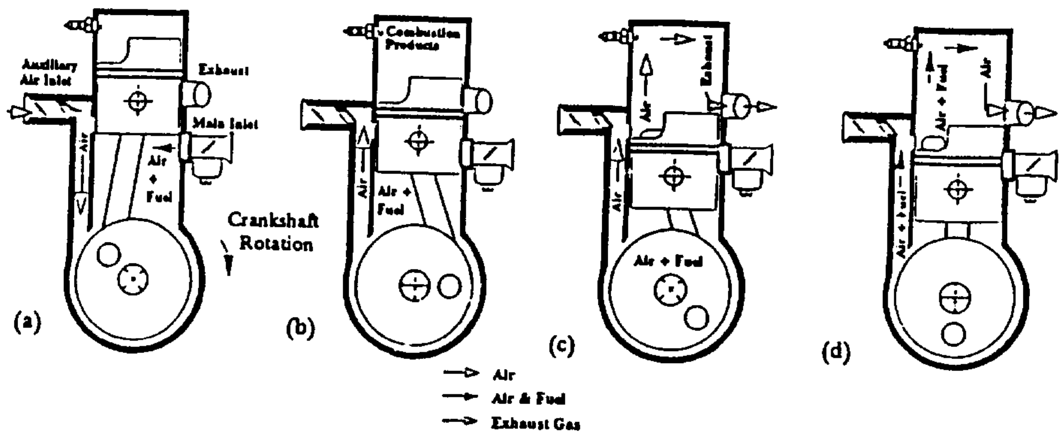


Figure 5: Operation of a two-stroke, loop scavenged engine.

As the piston gets closer to the bottom of its stroke, the transfer ports are uncovered, and air-fuel mixture from the crankcase is forced into the cylinder (Figure 5c). The pumping force required to move the air-fuel mixture is provided by the downward motion of the piston. Since the exhaust port is still open, the burned gases are pushed from the cylinder by the pressure of the incoming charge. In the process, however, some mixing between the exhaust gas and the charge takes place, so that some of the exhaust is retained in the cylinder, and some of the fresh charge is emitted in the exhaust. As the piston again rises for the next compression stroke, it closes first the transfer port and second the exhaust port - trapping the remaining charge in the cylinder. Before the exhaust port closes, however, the rising piston pushes some of the charge in the cylinder out into the exhaust (Figure 5d).

Since the gas exchange processes in a two-stroke engine are controlled by its piston and ports, the complex valve gear, camshaft, and related mechanisms needed in a four-stroke engine are not needed. For this reason, two-stroke engines are easier and cheaper to manufacture than four-stroke engines.

4.2 Causes of Emissions for Two-Stroke Engines

In small two-stroke utility engines, the major sources of unburned hydrocarbon emissions are the loss of unburned charge out the exhaust ports during scavenging, and hydrocarbon emissions due to misfire or partial combustion at light loads. The fraction of the total charge fed to the cylinder that is trapped to participate in the combustion process is known as the "trapping efficiency". At full load, trapping efficiency for a chainsaw engine may be as low as 55% (Blair, 1990) - implying that 45% of the fuel-air mixture supplied to the engine is emitted unburned in the exhaust.

Under light-load conditions such as idle, the flow of fresh charge is reduced, which increases the trapping efficiency. However, scavenging efficiency is also reduced, allowing substantial amounts of exhaust gas to be retained in the cylinder. This high fraction of residual gas can cause incomplete combustion or misfire. Misfiring or incomplete combustion cycles are the source of the "popping" sound commonly produced by two-stroke engines at idle and light loads, as well as the problems that these engines often have in maintaining stable idle. These unstable combustion phenomena are major sources of HC emissions under idle and light-load conditions.

Another source of the high HC and CO emissions typical of two-stroke engines is the air-fuel ratio, which is normally set very rich compared to (e.g.) four-stroke automotive engines. For conventional carburetted two-stroke chainsaw engines, the mixture is usually set around 12:1 by weight, compared to a stoichiometric air-fuel ratio of 14.7:1. This increases the maximum power output from the engine, and helps to limit the engine temperature, as well as providing easier starting. Since there is insufficient oxygen present in the cylinder to fully burn all the fuel to CO_2, however, substantial amounts of CO and HC are emitted in the exhaust.

Another source of HC emissions, as well as the high level of particulate emissions characteristic of two-strokes, is the lubricating oil that is added to the fuel to lubricate the crankcase parts. Since the crankcase is used as a pump, it cannot contain a pool of oil to lubricate the bearings as well. Thus, lubricating oil is mixed into the fuel instead. When the fuel is atomized in the carburetor and vaporizes, the less-volatile oil is left as a mist of oil droplets in the air-fuel mixture. Some of these droplets contact the cylinder walls, the crankshaft bearings, and other parts that require lubrication. Most of the oil, however, is carried into the combustion chamber along with the air-fuel mixture. The oil in that part of the air-fuel mixture that is not trapped and burned appears as particulate matter in the exhaust. Even the oil that is trapped often fails to burn completely. The presence of condensed oil droplets in the exhaust is responsible for the two-stroke's characteristic white or blue smoke emissions.

Since two-stroke SI engines usually retain significant exhaust gas in the cylinder and run at rich air-fuel ratios, flame temperature and NOx concentrations are usually low. Measures to reduce CO and HC emissions from two-strokes, to the extent that they result in leaner air-fuel ratios, are likely to increase NOx emissions.

4.3 Uncontrolled Emission Levels for Handheld Two-Stroke Engines

Since utility engines emissions have only recently been regulated, emission data are scarce, and are generally available only for new engines in proper running condition. The limited data available are presented in this section.

Figure 6 compares emissions from a number of uncontrolled two-stroke engines used in handheld utility equipment with the California 1995 standards. These data were compiled by the U.S. EPA. As expected, these data show very high HC and CO emissions. Many of these engines were calibrated to run at very rich fuel/air conditions for performance, stability and cooling

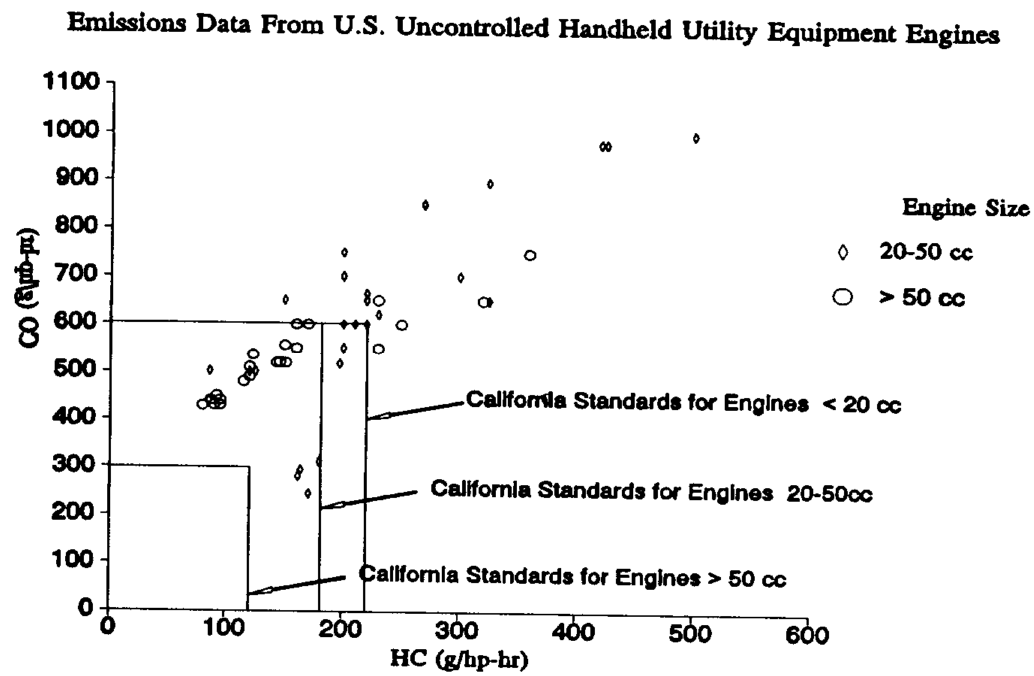


Figure 6: Emissions for uncontrolled handheld equipment engines.

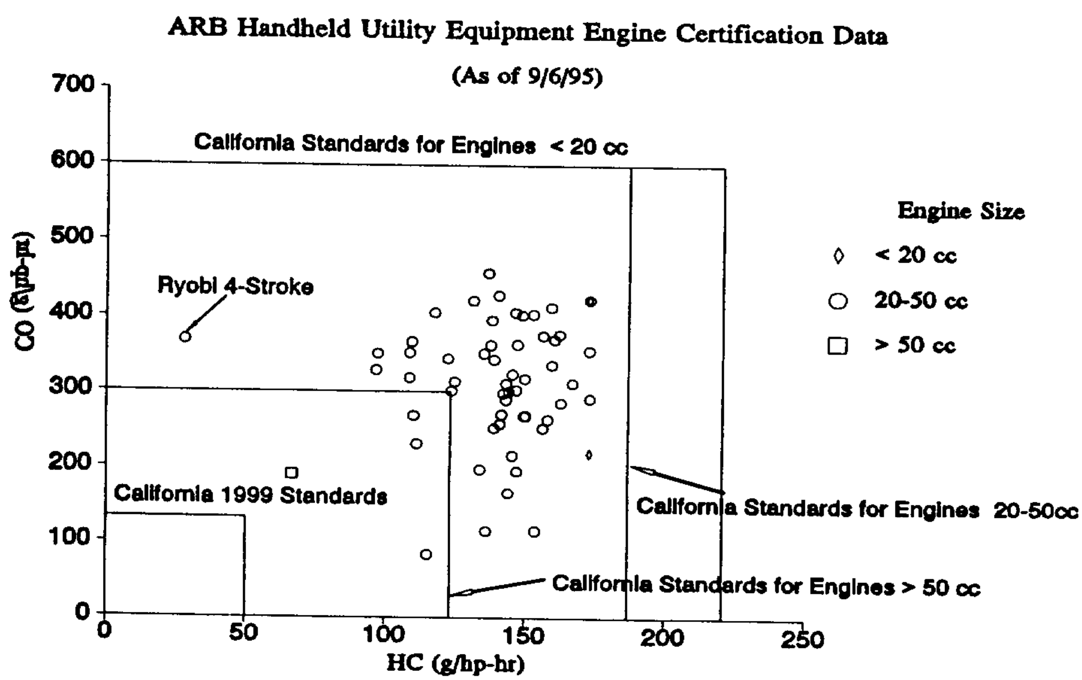


Figure 7: ARB certification data for 1995 handheld utility equipment engines.

purposes. However, Figure 6 also shows that a considerable number of engines in the 20 to 50 cc displacement range already met ARB's relatively lenient Tier I standards, even before such compliance became mandatory in 1995.

Table 4: Average emission levels for uncontrolled and controlled handheld equipment engines.

	Eng. Disp. (cc)	THC (g/bhp-hr)	CO (g/bhp-hr)	NOx (g/bhp-hr)
Engine Displacement: < 20 cc				
Uncontrolled	n/a	n/a	n/a	n/a
Controlled	18.0	163.6	314.7	0.82
Engine Displacement: 20 - 50 cc				
Uncontrolled	32.9	226.0	622.6	0.60
Controlled	31.8	138.9	312.2	1.19
Engine Displacement: > 50 cc				
Uncontrolled	63.3	159.6	518.6	0.20
Controlled	56.0	66.52	190.1	2.65

Figure 7 shows emissions certification data for California 1995 model handheld utility equipment engines. These engines were certified to California Tier I standards. The average emission levels for these certified engines and the uncontrolled engines are tabulated in Table 4. As this table shows, average HC and CO emission levels from uncontrolled utility engines have been reduced by about 40 to 60%. NOx emission levels increased noticeably due to the leaner air-fuel ratio, but are still much less than the HC and CO emissions. In addition, with emission standards in place, the emission levels show a much smaller range of variation compared to those of uncontrolled engines.

None of the 1995-model California-certified engines came close to meeting the Tier II emission standards. Although the HC emissions for the Ryobi four-stroke engine met the Tier II limit, the CO emission level was higher than the Tier II standard.

Another interesting observation on the emission data in Figure 7 is that the emission levels for the only engine with displacement greater than 50 cc were quite low. These data were for a 56 cc blower engine manufactured by Stihl. Discussions with a Stihl engineer revealed that the blower engine is designed to run leaner than other engines. This engine can afford to run leaner without fear of overheating, since the blower provides a very high flow of cooling air.

To meet the 1995 Tier I standards, most handheld equipment engines required only enleanment in fuel/air mixture, improvements in fuel metering, changes in ignition timing, and improved cooling to meet the relatively lenient Tier 1 emission standards. Some engines required minor design changes as well. However, as these tables and figures show, compliance with the Tier II emission standards will require major reductions in HC and CO emissions. To achieve these reductions will advanced engine modifications and/or use of aftertreatment devices.

4.4 Smoke and Particulate emissions

Data on PM emissions from small two-stroke utility engines are relatively limited. Tests carried out at Southwest Research have shown PM emissions from small two-stroke engines of the type used in chainsaws to be between about 4.0 and 8.0 g/bhp-hr, with larger engines tending to have lower brake-specific emissions (ARB, 1990; Hare and Carroll, 1993). To achieve the Tier II PM emission standard of 0.25 g/BHP-hr will thus require at least a 95% reduction in PM emissions from uncontrolled levels. However, in a presentation to the EPA, the Portable Power Equipment Manufacturers Association (PPEMA) reported that 0.63 g/bhp-hr of PM emissions was measured from a chainsaw engine with 2% oil mixture (50:1 oil/fuel ratio) and a lean air-fuel ratio of 16:1 (PPEM-A, 1994). When the oil/fuel ratio reduced to 100 to 1, the PM emissions reduced to 0.44 g/bhp-hr, which is still nearly twice the PM standard. For comparison, the oil/fuel and air/fuel ratios for existing handheld equipment engines are typically about 40 to 1 and 11 to 13:1, respectively. Meeting the PM emissions limit is expected to prove very challenging, and will probably require catalytic converters, major changes in lubrication systems, and/or use of reformulated lubricating oil, in addition to the technologies needed for HC and CO reduction.

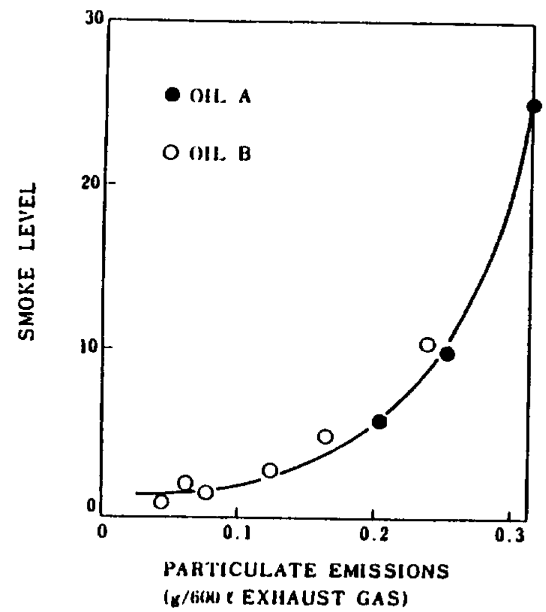


Figure 8: Correlation between smoke and PM emissions for two-stroke engines.

Data on the effectiveness of PM control measures for two-stroke engines are virtually nonexistent. The main exception is the work of Pfeiffer (1993), who found that a catalytic converter reduced PM emissions under most conditions by about 80%. Pfeiffer's data also show that visible smoke levels correspond fairly well with particulate emissions. Sugiura et al. (1977) also found a relationship between visible smoke and PM emissions in motorcycles (Figure 8). In reviewing motorcycle smoke and particulate emissions data collected by SwRI in 1973, we also concluded that the smoke and particulate emissions from two-stroke engines are correlated (Chan and Weaver, 1994). In a recent phone conversation, an engineer with Poulan observed that smoke and PM were also related in chainsaw engines (Liechty, 1995). Thus, techniques identified for reducing visible smoke will also help to reduce measured PM.

5. EMISSION CONTROL TECHNOLOGIES FOR SMALL TWO-STROKE ENGINES

As discussed in Chapter 4, the major sources of unburned hydrocarbon emissions in two-strokes are the loss of unburned air-fuel mixture out the exhaust valve during scavenging, and hydrocarbon emissions due to misfire or partial combustion at light loads. Losses due to "short-circuiting" of unburned charge into the exhaust can be as high as 45% for conventional two-stroke engines under full-load conditions (Hare et al, 1974; Batoni, 1978; Nuti and Martorano, 1985). Under light-load conditions such as idle, the flow of fresh charge is reduced, and substantial amounts of exhaust gas are retained in the cylinder. This high fraction of residual gas leads to incomplete combustion or misfire. Misfiring or incomplete combustion cycles are the source of the "popping" sound produced by two-stroke engines at idle and light loads, as well as the problems that these engines often have in maintaining stable idle. These unstable combustion phenomena are the causes unburned HC exhaust emissions at idle or light load conditions (Tsuchiya et al, 1983; Abraham and Prakash, 1992; Aoyama et al, 1977).

Technologies potentially applicable to reducing small two-stroke engine emissions can be grouped into the following categories:

- advanced fuel metering systems;
- improved scavenging characteristics;
- combustion chamber modifications;
- improved ignition systems;
- exhaust aftertreatment technologies;
- improvements in engine lubrication; and
- conversion to four-stroke engines.

The application of some of these technologies to small utility equipment, moped, and motorcycle engines to reduce exhaust emissions have been reported in a number of studies, and there is now significant practical experience with some of these techniques.

5.1 Advanced Fuel Metering Systems

Precise metering of air and fuel can improve engine performance and fuel consumption and reduce exhaust emissions. Conventional carburetion systems for small two-stroke engines are designed to provide smooth and stable operation under a variety of speed and load conditions, but give little consideration to fuel consumption or exhaust emissions. The potential advantages of fuel injection in two-stroke engines are two-fold: more precise control of the air-fuel ratio over the entire range of operation, permitting the engine to operate with leaner mixtures; and the possibility of using timed injection and/or in-cylinder injection to eliminate HC emissions due to short-circuiting of fresh charge during scavenging.

The electronic fuel injection systems used in modern automobiles provide a precisely metered amount of fuel, based on a measure of the air flow into the engine. The fuel supply system, which provides the fuel flow to the injection system, consists of a fuel pump, fuel filter and pressure regulator. The fuel injector is a high-speed solenoid valve connecting the pressurized fuel supply to the engine air intake. By opening the valve, the electronic control unit permits pressurized fuel to spray into the air intake, where it mixes with air, vaporizes, and is inducted into the engine.

A similar fuel injection system could be applied in advanced small two-stroke engines. This system could be configured to spray fuel either into the intake port, or into the crankcase, to provide better mixing and increased time for vaporization. Numerous studies have been undertaken with two-stroke engines using this approach to reduce exhaust emissions (Sato et al., 1987; Nuti, M., 1988; Plohberger et al., 1988; Beck et al., 1986; Duret et al., 1988; Huang et al., 1991; Leighton et al., 1994, Yoon et al., 1995).

By appropriate control of fuel injection timing, it is possible to reduce the hydrocarbon content of the air that short-circuits the combustion chamber during scavenging. Because of the need to assure a combustible mixture at the spark plug, however, it would not be possible to eliminate short-circuiting HC completely in port or crankcase-injected engines. Furthermore, some ingenuity is required to provide the pumping power necessary to maintain injection pressure without adding unduly to the size and cost of the engine. In this discussion, we will refer to this approach as "indirect injection".

An alternative fuel injection approach can eliminate short-circuiting entirely. This is to inject the fuel directly into the cylinder near or after the time that the exhaust port closes. This approach is generally referred as direct in-cylinder fuel injection. Because injection directly into the cylinder provides very little time for fuel mixing and vaporization, direct fuel injection systems must inject very quickly, and achieve very fine levels of atomization of the fuel. This type of fuel injection can use high-pressure, liquid-fuel injection systems to inject the fuel directly into the cylinder (Sato et al., 1987; Nuti, M., 1988; Plohberger et al., 1988; Beck et al., 1986; Yoon et al., 1995). Direct in-cylinder fuel injection can also be achieved with low-pressure fuel systems, using air-blast injection (Duret et al., 1988; Huang et al., 1991; Leighton et al., 1994; Yoon et al., 1995). For air-assisted direct-injection systems, an air pump or similar means is required to supply compressed air for the injection system.

The quality of the atomization of a fuel spray is usually measured in terms of the Sauter Mean Diameter (SMD) of the fuel droplets. In order to quickly vaporize the fuel spray, a fuel droplet SMD of 10 to 20 microns is usually required for direct (in-cylinder) fuel injection. For indirect injection, a fuel droplet SMD of 100 microns is quite acceptable, as the fuel will have time to vaporize in the intake port and during the compression stroke.

A good fuel-injection system needs to have the ability to deliver extremely small fuel droplet sizes, to control spray penetration and fuel distribution. It must also mix fuel adequately with all of the available air in the short time available at the high engine speeds typical of two-stroke operation. In addition to fine fuel droplet size, the fuel droplet size distribution must remain much the same throughout the fuel spray to assure minimum coalescence of the droplets towards the end of the spray plume.

Due to the achievements reported for engines using the air-assisted Orbital Combustion Process (OCP) and similar direct-injection approaches, two-stroke engines are presently a major area of automotive research and development. Some prototype two-stroke engines have reached emission levels comparable to good four-stroke engines. However, only limited studies have been carried out on the application of the advanced direct fuel-injection systems in small two-stroke engines such as those in motorcycles and small handheld utility equipment. The results of these studies are discussed in the following sections.

Institute Francais du Petrole

The Institute Francais du Petrole (IFP) designed and developed a marine outboard two-stroke engine based on a converted production engine, using a direct air-assisted fuel-injection system with compressed air supplied by the pumping action of the crankcase (Monnier and Duret, 1991). This system, named "IAPAC", includes a surge tank which stores the compressed air from the crankcase as shown in Figure 9. This surge tank serves as a reservoir to supply compressed air to the pneumatic fuel injection device.

Several versions of the IAPAC engine system were tested for performance and exhaust emissions, and the results are shown in Table 5. These included engines with the original and with a newly designed intake manifold, and an engine with external compressor for supplying the compressed air (named HIPAC). As shown in Table 5, IFP reported that this 1.2 liter, 85 hp engine was able to provide a reduction in fuel consumption of about 27%, and reductions in HC and CO of 84% and 79%, respectively. Although NOx emissions increased by 133%, these levels are still low compared to typical NOx emissions from four-stroke engines.

Recently, Piaggio of Italy has been working with IFP to incorporate the IAPAC into scooter engines (Monnier et al, 1993). As a result of this collaboration, a 125 cc Piaggio two-stroke engine equipped with the IAPAC compressed air assisted fuel injection system has been developed. According to Monnier et al, the Piaggio-IAPAC scooter produced 68% less HC+NOx emissions (from 6.60 g/km to 2.42 g/km), and about 87% less CO emissions (from 11.68 g/km to 1.47 g/km) on the ECE R40 driving cycle. It also consumed about 27% less fuel. With the use of a catalytic converter in this fuel-injection equipped scooter, the HC+NOx and CO were further reduced to 0.27 g/km and 0.05 g/km, respectively. However, a 10% reduction in vehicle power was observed due to the use of the catalytic converter, presumably due to its effect on gas-dynamics in the exhaust. This loss could be overcome by re-optimizing the exhaust

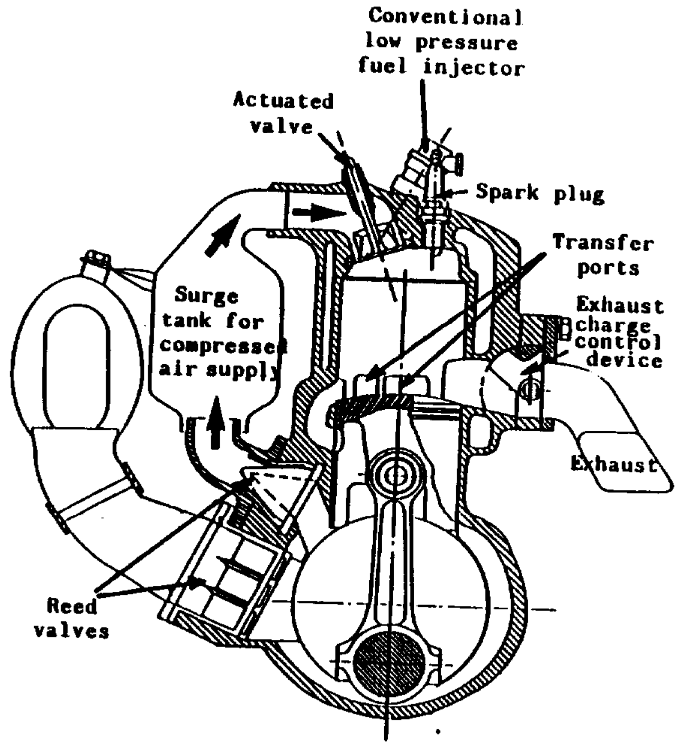


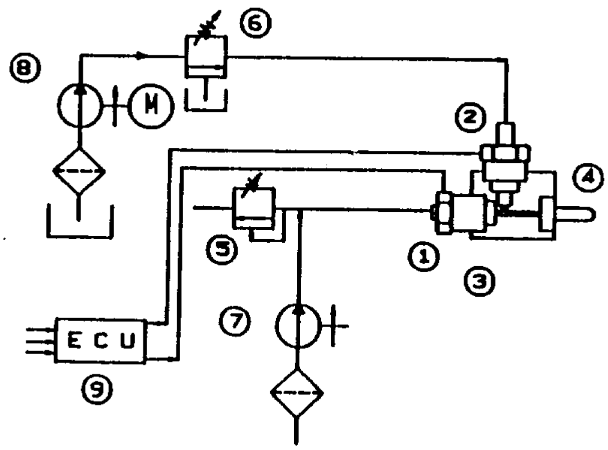
Figure 9: Schematic of a two-stroke engine with the IFP IAPAC system.

Table 5: Engine performance and exhaust emissions data for modified marine two-stroke engine designed by IFP.

System Configuration	Emissions (g/Kw-h)			BSFC (g/Kw-h)	Power (Kw)
	HC	CO	NOx		
Production	164	211	1.5	514	59.2
IAPAC w/ original intake system	55	69.9	3.5	396	56.8
IAPAC w/ new intake system	44	61.5	3.5	365	56.8
IAPAC w/ new intake system and external compressed air (HIPAC)	26	43.9	3.5	373	56.8

system. Monnier et al also stated that a low cost electronic control unit has been developed, with the goal of commercializing the technology for scooter and motorcycle applications.

IFP is also working on a slightly different version of its IAPAC system named ROTAPAC (Glover and Duret, 1993). The main different between the IAPAC and ROTAPAC systems is



1. air injector	5. air pressure regulator	9. ECU
2. fuel injector	6. fuel pressure injector	
3. adapter	7. air pump	
4. nozzle	8. fuel pump	

Figure 10: Schematic of the ITRI air-assisted fuel injection system.

that the ROTAPAC system uses a rotary valve, instead of a poppet valve, as a means of introducing the fuel/air mixture into the cylinder. According to Glover and Duret, some preliminary results from a 125 cc two-stroke engine equipped with the ROTAPAC system showed significant fuel consumption and HC emissions reductions as compared to conventional two-stroke engines.

<u>Industrial Technology Research Institute</u>

Researchers at the Industrial Technology Research Institute (ITRI) of Taiwan have successfully demonstrated the use of a low-pressure, air-assisted, direct fuel-injection system in a two-stroke scooter engine (Huang et al, 1991; 1993). The schematic of the air-assisted fuel-injection system is shown in Figure 10. Prior to the development of this air-assisted cylinder head fuel-injection system, ITRI tested a cylinder-wall fuel-injection system with and without air assist for fuel atomization. Table 6 shows the average dynamometer exhaust emissions and fuel economy results for two 82 cc two-stroke scooters equipped with stock carburetted engines under best-tuned conditions, and compares these with those of scooters equipped with different fuel-injection system designs. Table 6 also shows the result of equipping the scooter engines with the air-assisted fuel injection in the cylinder head, along with skip-injection control and a catalytic converter. These technologies will be discussed later in the section.

The ITRI researchers achieved a fuel droplet SMD of 70 microns for the cylinder wall injection system without air assist. This was done by introducing an injector nozzle with spiral grooves. As shown in Table 6, reductions of 25% in HC and 8% in CO emissions, as well as a 14% improvement in fuel economy were achieved with this approach. As expected, the NOx emissions increased substantially due to the leaner combustion.

Table 6: Dynamometer exhaust emissions and fuel economy data for a scooter engine equipped with different fuel-injection system designs developed by ITRI.

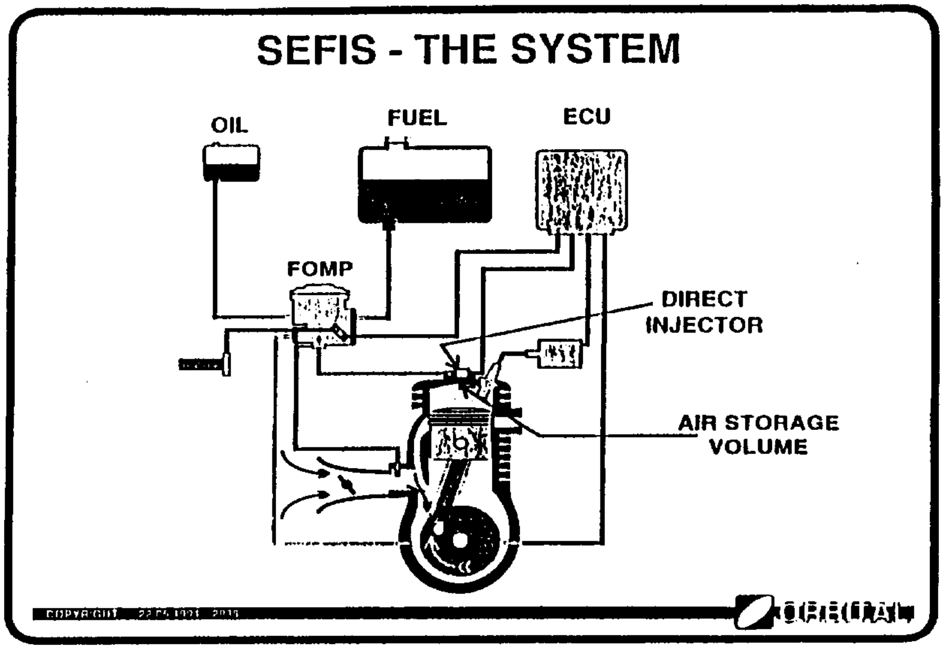
System Configuration	Emissions (g/km)			Fuel Econ. (km/l)
	HC	CO	NOx	
Carburetor System	3.83	3.7	0.03	42.1
Cylinder wall injector	2.90	3.4	0.06	48.0
Air-assisted cylinder wall injector	2.58	3.0	0.09	50.4
Air-assisted cylinder head injector	2.12	2.5	0.16	52.8
Air-assisted cylinder head injector with skip-injection	1.62	1.6	0.17	55.1
Air-assisted cylinder head injector with skip-injection and catalytic converter	0.28	0.09	0.16	55.3
Air-assisted cylinder head injector with skip-injection, and catalytic converter with secondary air	0.25	0.08	0.16	55.0

With the use of low-pressure air assist in the cylinder-wall injection system, the fuel droplet SMD was decreased to 35 microns. Compared to the carburetor system, this air-assisted fuel-injection system reduced the HC and CO emissions by 34% and 18% respectively, and improved the fuel economy by 20%. NOx emissions increased by 200%, however. Even greater reductions in HC and CO emissions and further improvement of fuel economy were achieved with the air-assisted fuel-injection system in the cylinder head. This produced a 45% reduction in HC and 32% reduction in CO emissions, along with a 53% improvement in fuel economy. This large effect is presumably due to the better charge stratification possible with the fuel injection system in the cylinder head. The SMD of the fuel droplets was reduced to 15 microns with this system.

In mass production, the ITRI direct fuel injection system would weigh about two pounds, according to ITRI staff, excluding the power source for the system. However, they estimated the weight of the system could be reduced to about one pound for small engine applications.

Orbital Engine Co.

For several years, Orbital has been refining its Orbital combustion process (OCP) for small two-stroke engine applications (Leighton et al, 1994-2). Recently, a small engine fuel injection system (SEFIS) utilizing the OCP principle has been developed. A schematic of the SEFIS system is shown in Figure 11. According to Leighton et al, the SEFIS is a low-cost fuel injection system which aims to share most of the hardware with systems intended for automotive applications. One of the distinctive features that Orbital claims for the SEFIS system is that it does not require a separate air compressor. The compressed air needed for the air-assisted fuel injection system is obtained from the captured gas in the cylinder during the compression stroke. Fuel



Source: Leighton, (1994-2)
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Figure 11: Schematic of the Orbital SEFIS air-assisted fuel injection system.

Table 7: Emission data from motorcycles using Orbital SEFIS system.

Emissions from a 50 cc Scooter Using Orbital SEFIS on ECE 40 Driving Cycle (g/km)				
Emissions	**Baseline**	**SEFIS w/o Throttle**	**SEFIS w/ Throttle**	**SEFIS w/ Catalyst**
HC	4.30	0.94	0.56	0.11
CO	5.00	1.29	0.47	0.01
HC+NOx	4.30	1.15	0.83	0.39
Emissions from a 150 cc Motorbike Using Orbital SEFIS on ECE 40 Driving Cycle (g/km)				
HC	3.80	n/a	0.58	n/a
CO	2.40	n/a	0.64	n/a
HC+NOx	3.81	n/a	0.75	n/a

pressurization and the separate oil metering system are powered by crankcase pressure fluctuations. Some emissions data from a scooter and a motorcycle equipped with the Orbital SEFIS are shown in Table 7. These emission results show substantial reductions in HC+NOx and CO emissions when the Orbital SEFIS was used. Further emission reductions were also achieved when a catalyst was used along with the SEFIS.

It has been reported that Mercury Marine is using the SEFIS for some of its outboard applications, and commercial outboard engines equipped with the SEFIS are scheduled to be launched in 1996. Recently, conversation with several Orbital staff revealed that Orbital is in discussions with several scooter/motorcycle manufacturers in Taiwan, as well as in China, to develop fuel-injected scooters or motorcycles using the Orbital SEFIS.

FICHT

Ficht GmbH, a research and development company in Germany, is also developing a high pressure injection system named DSE-PDS for two-stroke engines (Heimberg, 1993). A schematic of Ficht's DSE-PDS is shown in Figure 12. Unlike the air-assisted types, the Ficht DSE-PDS injection system uses the "water hammer" principle to transform kinetic energy to a pressure pulse which atomizes and injects the fuel. Ficht reports that the system has been and is being tested by various two-stroke as well as four-stroke engine manufacturers with engine sizes ranging from 50 to 500 cc, including some makers of scooter and outboard engines. Compared to carbureted two-stroke engines, Ficht claims that two-stroke engines with the DSE-PDS injection system can reduce HC emissions up to 90% and CO emissions up to 75%. While mass emission data for a 50 cc two-stroke engine with Ficht at full load condition were presented, no emission results were given under low or light load condition in the literature.

In 1992, Outboard Marine Corporation (OMC) acquired from Ficht GmbH an exclusive license to use the Ficht DSE-PDS injection system in the marine sector. Jointly with OMC, Ficht has prepared a variety of marine engines equipped with the DSE-PDS system for evaluation at OMC. A recent conversation with Ficht representatives indicated that there are some discussions with small engine manufacturers to explore the possibility of using the DSE-PDS system for small engine applications.

Prototype Injection Systems for Handheld Equipment Engines

In addition to the IFP/Piaggio, ITRI, and Orbital studies, there are a number of unpublished studies underway to develop low cost fuel injection systems for handheld equipment engines, such as those for chainsaws. Other organizations known to be working on these types of systems include BKM in San Diego, Fuel Management Systems (FMS) in Illinois, and Stihl.

BKM - The BKM project is partly funded by the New York State Energy Research and Development Authority. As of today, BKM has demonstrated a "proof of concept" breadboard prototype chainsaw equipped with its Servojet high pressure, liquid-fuel direct injection system. In its demonstration, the auxiliary parts that are required to run the fuel injection system, such as fuel and oil pumps, electric motors, pressure regulators etc., were not integrated with the chainsaw. The BKM breadboard prototype has been tested at the University of Michigan, with the injection system operated from a standard 120 volt electrical supply. The average emission data with two different injectors and cylinder heads were reported as 22.4, 0.96 and 70 g/hp-hr

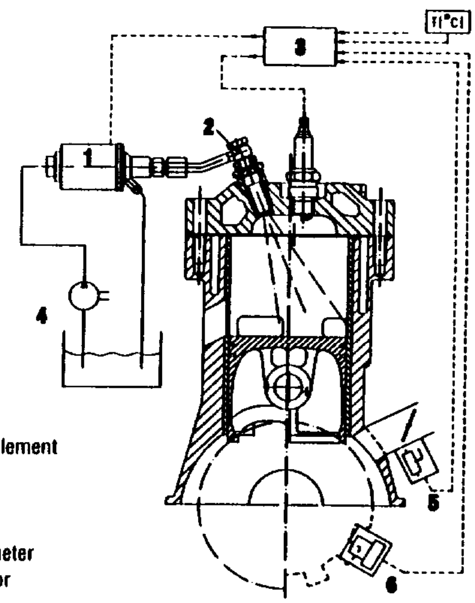

Figure 12: Schematic of a two-stroke engine with the Ficht DSE-PDS system.

for HC, NOx, and CO emissions, respectively (EPA, 1995), and no PM emission results were reported. The PM emissions is expected to be quite similar to uncontrolled levels, since no special steps were taken to reduce emissions of unburned lubricating oil. Although the test results demonstrated that the breadboard prototype could meet ARB's Tier II standards, the major problems in incorporating the system into a self-contained, portable chainsaw remain to be resolved.

FMS - The FMS project was funded by the Swiss Department of Forestry. A prototype indirect (intake port) fuel injection chainsaw, has been developed and tested at the Swiss Federal National Test Institute. This system used an FMS electronic control unit and Siemens automotive fuel injector. It is claimed that on the GUT cycle, 33% and 85% reductions in HC and CO emissions were achieved, with a 408% increase in NOx emissions. A video provided by the FMS also documents the laboratory and field testing of the prototype chainsaw. However, similar to the BKM prototype, the system auxiliaries required to run the prototype chainsaw were not integrated or built into the chainsaw, but were powered by a separate car battery.

Stihl - Stihl, a major manufacturer of handheld equipment, is developing a prototype mechanical direct fuel injection chainsaw. A schematic of Stihl's mechanical direct fuel injection system is shown in Figure 13. For this prototype, all of the components have been integrated into the chainsaw, without auxiliary components such as an external energy supply. As the fuel and lubricating oil are supplied to the engine separately, the prototype chainsaw is equipped with a lubricating oil system, comprising an oil tank, filter, pump, and injection channel. This is different from the current fuel/oil mixing system used in carbureted chainsaws. Similar to the Orbital SEFIS system, the Stihl injection system also uses the pressure pulses of the crankcase

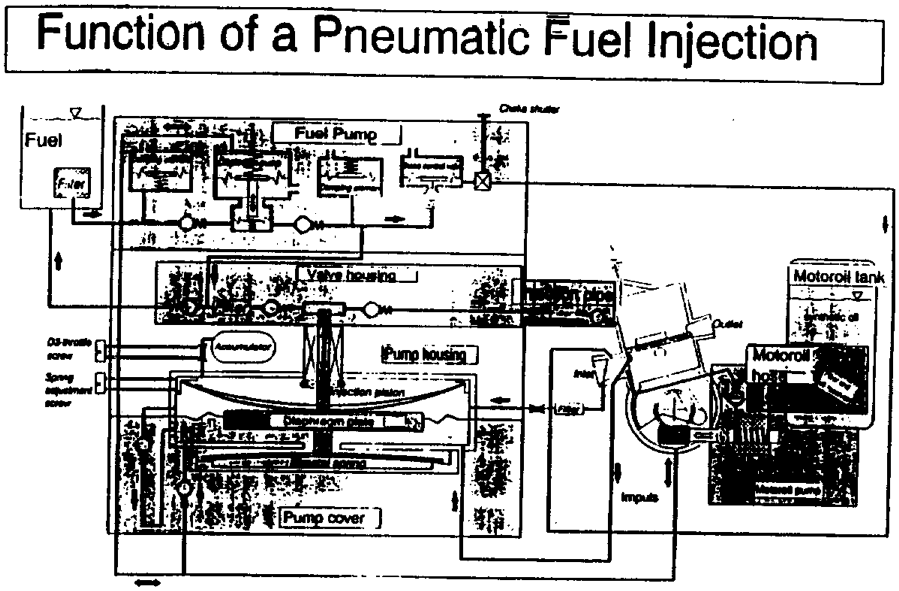

Figure 13: Schematic of the Stihl mechanical direct fuel injection system.

to drive the fuel pump. The Stihl fuel injection system does not use any electronic control system, and the injection timing is controlled by a hole in the piston skirt. The emission results for this prototype chainsaw were reported as 20 g/hp-hr for HC emissions, and 200 g/hp-hr of CO emissions. Thus, the prototype would meet the Tier II HC emission standard, but not the CO standard. According to a Stihl engineer, a few of these prototype chainsaws have been evaluated in the field, and the results were encouraging. Because of the additional parts, especially costly precision parts, required, the cost of these chainsaws would be significantly higher than for present units. Stihl has estimated that the incremental cost to the customer for these advanced chainsaws would be $200 per unit, even at high production volume.

All these data and studies, as well as data from other studies of fuel-injection systems for automotive two-stroke engines, show that significant reductions in HC and CO emissions can be achieved through the use of fuel-injection systems. However, innovative design will be needed to develop a practical, economical, and efficient fuel injection system that can be installed in a two-stroke engine for handheld equipment.

5.2 Skip-firing

Besides precision in fuel metering, another advantage of electronically controlled fuel-injection systems is the ability to shut off fuel injection in some engine cycles. With this feature, the fuel supply can be shut off for a definite number of consecutive engine cycles under idle and light engine load conditions. These are the conditions at which misfire and irregular combustion usually occur. This allows time for exhaust gas to be purged from the combustion chamber, and thus providing better combustion conditions for the next designated engine firing cycle. Thus, irregular combustion under light engine load conditions can be eliminated or minimized.

Researchers at ITRI have successfully applied this skip-injection technique to a scooter engine to minimize unburned HC emissions due to irregular combustion under idle and light load conditions (Huang et al, 1992 & 1993). ITRI researchers found that, without skip-firing, the indicated mean effective pressure (IMEP) varied significantly at idle, and many cycles could easily be identified as having incomplete combustion cycles or even complete misfire. This resulted in a very high concentration of unburned HC emissions: of the magnitude of 3,500 to 4,000 ppm of hexane equivalent in the exhaust. Several skip-injection modes were investigated, including fuel injection every other cycle, and every three, four and five cycles. The results showed that IMEP variations decreased as the number of skipped injections increased. In an engine dynamometer test with fuel injected every four cycles, the HC emissions and fuel flow rate at idle were reduced by 50% and 30% respectively. This skip-injection mode was also applied and tested in a scooter engine, producing the exhaust emissions and fuel economy results shown earlier in Table 6. Reductions of HC and CO emissions of 58% and 57%, respectively, and a 31% improvement in fuel economy were demonstrated with this approach.

It has been reported that the BKM high pressure fuel injection system for two-stroke engines also uses this approach during idle and low load conditions.

5.3 Scavenging Control Technologies

In a two-stroke engine, the exhaust and intake events overlap extensively, as the piston finishes its downward stroke and begins its movement from the bottom of the cylinder to the top. As the piston approaches the bottom of the cylinder, exhaust ports in the walls of the cylinder are uncovered. The high pressure combustion gases blow into the exhaust manifold. As the piston gets closer to the bottom of its stroke, the intake ports are opened and fresh air or air-fuel mixture is blown into the cylinder while the exhaust ports are still open. Piston movement timing (measured in crank angle) and cylinder port configuration are the major factors controlling the scavenging process. The ideal situation would be to retain all of the fresh charge in the cylinder (high trapping efficiency) while exhausting all of the spent charge from the last cycle (high scavenging efficiency). These two goals conflict. In production engines, the cylinder ports and timing are generally designed for high scavenging efficiency, in order to achieve maximum power output and smoother idle, at the expense of higher short-circuiting losses and HC emissions. It is possible to reconfigure the intake and exhaust ports to fine-tune the scavenging characteristics for lower emissions, but this involves significant trade-offs with engine performance. Another way to increase trapping efficiency, with minimum impact on performance, is to apply exhaust charge control technology.

Exhaust charge control technology modifies the exhaust flow by introducing one-way control valves in the exhaust, or by making use of the exhaust pressure pulse wave. Using the exhaust pressure pulse wave to control intake and exhaust flow usually requires a fairly long exhaust pipe, and is effective only for a restricted range of engine RPM. For this reason, one-way control valves are usually used to control the exhaust flow rate in small engines. The critical variable parameter for exhaust charge control techniques is the contraction ratio, which is defined as the ratio of the restricted exhaust passage area regulated by the valve to the unrestricted exhaust passage area. The effectiveness of these techniques is measured by the delivery ratio, which is the ratio between the mass of air-fuel mixture actually delivered to the engine and the mass of air-fuel mixture contained by the engine displacement volume at ambient conditions.

Exhaust Control Valve

Hsieh et al (1992), Tsuchiya et al (1980), and Duret and Moreau (1990) have demonstrated the potential of exhaust charge control valves in small two-stroke engines. Results of their studies show that significant reductions in HC emissions and fuel consumption can be achieved, as well as a reduction in unstable combustion at light load. A study done by Yamagishi et al. (1972) concluded that misfire is most likely to occur at a delivery ratio less than 0.3. It was also observed that the scavenging losses were low but the exhaust HC concentration was still high. At a low delivery ratio, the trapping efficiency was higher and resulted in lower scavenging losses. However, at very low delivery ratios corresponding to light load and idle, misfire or irregular combustion were occurring - resulting in high HC emissions even though the scavenging losses were low.

Tsuchiya et al. identified the delivery ratio at which a rapid increase in irregular combustion occurs (defined as the critical delivery ratio) as 0.2. This is very similar to the finding of Hsieh et al that 0.25 was the critical delivery ratio. With exhaust charge control, Hsieh et al. found that the critical delivery ratio decreases from 0.25 to 0.20 and 0.15 at low and medium engine speeds (1,500 and 3,000 rpm), respectively. Thus, the exhaust charge control technique effectively reduced irregular combustion under light-load conditions. Hsieh et al. found that HC emissions and fuel consumption were reduced by 30% and 6% respectively when the exhaust charge control technique was used in a test engine. Also, at the same delivery ratio, the engine with exhaust charge control produced higher power output. Duret and Moreau found that a 60% reduction in HC emissions and 20% reduction in fuel consumption could be achieved through the use of an exhaust charge control valve.

Honda has incorporated a "Revolutionary Controlled Exhaust Valve (RC Valve)" in a 150 cc two-stroke motorcycle model equipped with a capacitive-discharge ignition, computerized controller and servo motor to attain high power efficiency at low and high speed conditions. Although the "RC Valve" is intended to improve engine performance, it can also serve as an emissions control device.

Stratified Scavenging

Stratified-charge two-stroke engines with fuel injection systems include the OCP, PROCO and DISC engines developed by Orbital, Ford and General Motors, respectively. These have been shown to reduce or eliminate scavenging losses. However, the cost-effectiveness and practicality of using these technologies in small two-stroke engines are still open to debate.

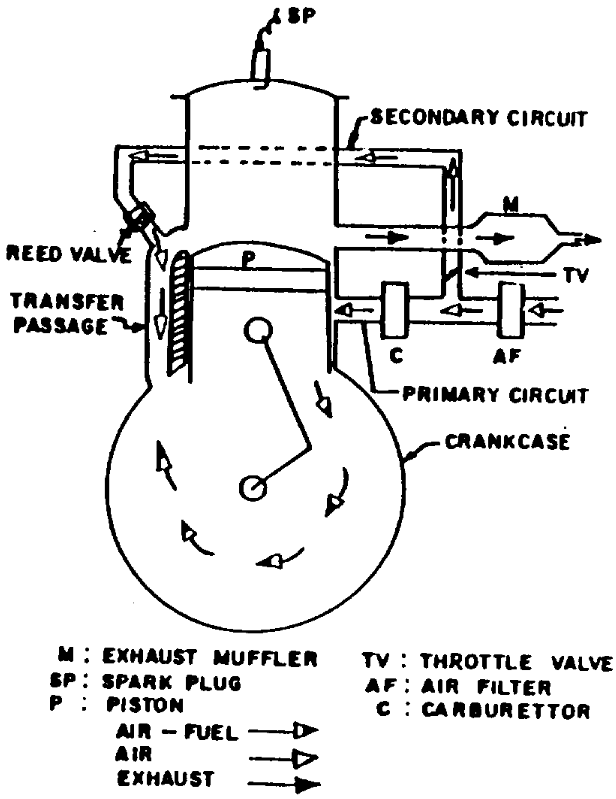

Figure 14: Schematic to illustrate the stratified scavenging approach in a two-stroke engine.

Another interesting concept, which is much simpler and less costly than the fuel-injection systems, is the two-step stratified scavenging approach to reduce scavenging losses (losses due to short-circuiting), and hence the HC emissions. An engine configurations using this scavenging approach is shown in Figure 14. In this approach, a supply of pure air is first introduced into the cylinder during the start of the scavenging process, to displace the exhaust gas, and is then followed later by a rich fuel/air mixture to support combustion. Controlled by reed valves, the secondary air supply can be inducted either through transfer passages or through the crankcase. The simplest approach is to pre-fill the transfer passages between the crankcase and combustion chamber with pure air, inducted separately from the air-fuel mixture entering the crankcase.

Ideally, with stratified scavenging, most of the charge lost from the cylinder will be pure air or a very lean air-fuel mixture. Thus, scavenging losses are minimized. However, it is impossible to obtain a perfect "two-layer" charge (pure air, and rich fuel/air mixture), while accurately supplying the right amount of pure air to effectively displace the exhaust gas during the actual scavenging process at different load/speed conditions. Thus, there will still be some fresh fuel/air charge scavenged out the exhaust port(s) with the stratified scavenging approach. A number of researchers have tested different designs based on the stratified scavenging concept, and their designs are discussed in the following paragraphs.

Queen's University of Belfast - At the Queen's University of Belfast (QUB), Magee et al. have developed an "air head" scavenging system that uses the stratified scavenging concept in a 50 cc two-stroke engine (Magee et al., 1993). In this engine, two charge inlets are used: one for pure air only, and the other for the regular carburetion intake. Controlled by reed valves, the pure air is inducted into the top of the transfer passages through an auxiliary air inlet, while a mixture of air and fuel is inducted into the crankcase. At wide open throttle and maximum secondary air flow, Magee et al reported a 30% reduction in HC emissions and a 10% improvement in brake specific fuel consumption throughout the speed range. The fuel trapping efficiency for the engine with the stratified scavenging system was improved by 10%, and the performance (power and BMEP) of the engine also improved. However, in another paper on the same study (Magee et al., 1993-2), it was reported that the engine experienced some bad performance characteristics at light and part-load conditions, with BMEP less than 3.5 bar. It was suspected that this was due to the low delivery ratio, which was around 0.25-0.35 (Magee et al, 1993-2).

Ricardo - Recently, Ricardo Consulting Engineers PLC published a paper discussing their stratified charging concept (RSCE) (Glover and Mason, 1995). This system employs quite similar principles to the QUB's concept. In the Ricardo system, the fuel is initially delivered to a specially-shaped rear transfer port, which serves as a storage as well as a fuel preparation area. During the initial of the scavenging process, pure air is driven from the crankcase to the combustion chamber through two lateral transfer ports. Controlled by the piston, the rear transfer port is opened almost at the end of the scavenging process. This allows some remaining pure air to flow through the port and mix with the fuel, carrying the fuel/air mixture into the combustion chamber. Ricardo has demonstrated this concept in a 50 cc scooter engine. Emission results indicate that substantial HC emission reduction is achieved only during medium/high engine speed/load conditions. The stratified charging engine was found to be more unstable than the baseline engine, and it produced as much HC emissions as well.

Indian Institute of Petroleum - Saxena et al. (1989) of the Indian Institute of Petroleum have developed a 150 cc engine using the stratified scavenging concept with a dual intake system. The secondary pure air is induced into the transfer passages through reed valves. The primary and secondary air supplies were chosen to be 50% each, as Saxena et al's experiment showed that the HC emission reductions and BSFC level off after the supply of secondary air exceeded 50%. At full load conditions, the results showed 25 to 30% reductions in HC emissions, as well as about 10% improvement in BSFC throughout the range of air/fuel ratios tested (0.75-1.05). The performance of the engine was also improved slightly. Saxena et al. also showed the effect engine load on HC emissions and BSFC. Lower HC emission reductions (13-16%) and BSFC (2-3%) were found at low-load conditions, while at high load the benefits were a more than 30% reduction in HC emissions and a 10% improvement in BSFC. Under simulated road-load conditions (over a range of speeds) at an air/fuel equivalence ratio of 0.85, the HC emission reductions varied from 20 to 30% and BSFC improvements varied from 5 to 10%, depending on the engine speeds. Again, lower HC reductions and BSFC improvements were found at lower road/speed conditions.

To explain the low HC emission reduction and BSFC improvement at low load conditions, Saxena et al. determined the fuel trapping efficiency and scavenging losses from both engines for a range of delivery ratios (0.2-0.6). High fuel trapping efficiencies and low scavenging losses were observed at low delivery ratio (light load conditions), similar to Yamagishi's findings; and low fuel trapping efficiency and high scavenging losses were found at higher delivery ratios. Compared with the base engine, the fuel trapping efficiency was improved and the scavenging losses were decreased for the stratified scavenging engine throughout the range of delivery ratios. However, minimal improvements in both fuel trapping efficiency and scavenging losses were achieved at low delivery ratios or light load conditions. With these results, Saxena et al. concluded that at low delivery ratio the losses due to poor combustion were high and the scavenging-through losses were low, and therefore, minimal HC emission reduction could be achieved with this dual-intake stratified scavenging engine.

India Institute of Technology - Babu et al. (1993), of the Indian Institute of Technology, also investigated the stratified scavenging approach. Similar to the system investigated by Saxena et al, Babu et al. used a second intake system to induct pure air through reed valves into the transfer passages. A difference was that the secondary intake system was set up to be able to supply compressed air. The system was applied on three engines, with engine displacements ranging from 55 cc to 250 cc. A control valve was used in the secondary air intake system to regulate or vary the air flow through the intakes. A compressor was set up to supply a slightly higher pressure air flow through the reed valves if needed. Also, three openings were selected to regulate the air flow through the reed valves to determine the effect of the amount of secondary air induced into the engine. In general, the results showed reductions in HC emissions of about 25%, and as much as 17% improvement in the brake thermal efficiency due to the reduction in scavenging losses. When an optimum amount of compressed air was supplied to the secondary air intake, the improvement in brake thermal efficiency and reduction in HC emission were even higher; especially at the full throttle condition. This was mainly due to the reduction in secondary air flows during high throttle conditions when the secondary air was induced at the atmospheric pressure. In this study, it was also concluded that an optimum secondary air flow rate was necessary in order to obtain maximum performance and emission benefits.

5.4 Other Engine Modifications

Other engine modifications and techniques include improved combustion chamber and piston configurations, improved lean/dilute combustion to prevent misfire, and changes in port and ignition timings.

Combustion Chamber - Combustion chamber and piston configurations can be improved to induce more turbulent motion to improve mixing during the compression stroke, as well as to control the flow direction of the fresh charge to minimize short-circuiting. Using improved combustion chamber and piston configurations with more swirl and squish can also minimize the formation of pocket or dead zones in the cylinder volume where burned gases can become trapped and escape displacement or entrainment by the fresh scavenging flow. Laimbock et al.

of Graz University of Technology (GUT) in Austria have designed a "jockey-cap" shape like combustion chamber which concentrated the squish area only above the exhaust port (Laimbock and Landerl, 1990). This "jockey-cap" type combustion chamber is designed to force the fresh charge to overflow the spark plug, which improves the cooling and allows the engine to run leaner without pre-ignition. Recently, Kawasaki modified the combustion chamber [illegible] the

exhaust timing for HC and NOx emission reductions.

Ignition Timing - The effect of ignition timing on two-stroke engines is essentially the same as on four-stroke engines. Retarding ignition timing beyond the minimum for best torque (MBT) point reduces power and increases fuel consumption, but reduces NOx and (within limits) HC emissions. Retarding ignition timing, especially at high loads, may offer a means of recovering much of the increase in NOx emission that will otherwise result from using a leaner mixture in low-emission two-stroke engines. Advancing ignition timing at light load reduces HC emissions in direct fuel-injected engines by reducing the dispersion of the fuel cloud. The cloud is therefore less likely to contact the walls of the combustion chamber. This reduces the amount of unburned HC produced by the quenching effect at the combustion chamber walls, as well as the filling of crevice volumes with unburned mixture. The unburned HC due to flame quenching and crevice volumes are major sources of HC exhaust emissions. With better combustion quality at advanced ignition timing, CO emissions are also reduced. NOx emissions, however, are increased with advanced ignition timing.

Dual Spark Plug Ignition - Researchers at ITRI have used a dual spark plug ignition in a scooter two-stroke engine to determine the effects on engine torque and unburned HC emissions (Huang et al, 1991). It was reported that the engine with dual spark plugs yielded lower HC emissions and better engine torque at low and medium engine load conditions. The improvements were believed to be due to the increase in combustion speed and the decrease in mixture bulk quenching effect when the dual plug ignition was used. However, ITRI's findings also showed that using additional spark plugs did not improve the high unburned HC emissions under idling or light-load conditions.

Recently, Kawasaki has investigated some traditional engine modifications to control emissions from a 25 cc chainsaw engine (Tamba S. et al, 1995). The findings of the study showed that 48% HC emission reduction and 85% CO emission reduction were achieved through enleanment and retarding exhaust timing. Some other minor modifications, such as combustion chamber and exhaust port modifications, and improved cooling ability, were also performed to the engine to overcome engine and exhaust gas temperature rise due to the leaner mixture, and power loss problems. Although the emission results from this modified engine did not meet the ARB Tier II emission standards, the study indicates that there is still a lot of room for engine modifications for small utility engines to reduce emissions. More research work should be encouraged on utilizing these more traditional emission control technologies to explore the extent of emission reductions that could be achieved through these technologies.

5.5 Exhaust Aftertreatment Technologies

The use of aftertreatment technologies such as thermal oxidation and catalytic converters can provide additional control of emissions beyond that achievable with engine and fuel-metering technologies alone. Catalytic converters are used extensively in automobiles, and have also been demonstrated on a limited basis in small two-stroke engines.

Thermal Oxidation

Thermal oxidation is used to reduce emissions of HC and CO by promoting further oxidation of these species in the exhaust. This further oxidation usually takes place in the exhaust port or pipe, and may require the injection of additional air to supply the needed oxygen. Substantial reductions in HC and CO emissions can be achieved through thermal oxidation if the exhaust can be maintained at a high enough temperature long enough. The typical temperature levels required for HC and CO oxidation are about 600 and 700 °C respectively. Although these requirements are difficult to meet for small engines with typical short exhaust systems, the technique has been demonstrated in a modified small four-stroke engine by introducing secondary air into the stock exhaust manifold upstream of the engine muffler. Air injection at low rates into the stock exhaust system was found to reduce emissions by as much as 77% for HC and 64% for CO (White et al, 1991). However, this was effective only under high-power operating conditions. In addition, the high exhaust temperatures required to achieve this oxidation would substantially increase the skin temperature of the exhaust pipe.

Oxidation Catalyst

Like thermal oxidation, the oxidation catalyst is used to promote further oxidation of HC and CO emissions in the exhaust stream, and it also requires sufficient oxygen for the reaction to take place. Some of the requirements for a catalytic converter to be used in two-stroke engines include high HC conversion efficiency, resistance to thermal damage, resistance to poisoning by sulfur and phosphorus compounds in the lubricating oil, and low light-off temperature. Additional requirements for catalysts to be used in two-stroke utility engines include extreme vibration resistance, compactness, and light weight.

Catalytic converters available for small utility engines use either metal or ceramic substrates. Although metal substrates have many advantages - especially resistance to vibration and shock - they are considerably more expensive. "Ballpark" pricing data from industry sources suggest that a catalytic converter based on a ceramic substrate should cost the engine manufacturer about five dollars per unit, while the cost of a metal substrate would be about four times as high. Although most available test data are on metal substrate catalysts, ceramic substrate suppliers have developed mounting techniques which they believe will allow catalytic converters using these substrates to give durability and performance similar to those of the metal substrate units. Recently, Corning and Engelhard have presented results indicating that the durability requirements of two-stroke engine can be met with the ceramic catalyst substrates with improved mounting systems (Reddy et al., 1995).

Application of catalytic converters to two-stroke engines presents a problem, because of the high concentrations of HC and CO in their exhaust. If combined with sufficient air, these high pollutant concentrations result in catalyst temperatures that can easily exceed the temperature limits of the catalyst. These high temperatures also pose a hazard of fire or personal injury to utility equipment users. Temperature limits for catalytic converters are similar for metal substrate and ceramic substrate catalysts - both begin to suffer damage at about 1000 °C. Thus, application of catalytic converters to two-stroke engines requires either limiting the air supply to limit pollutant oxidation and the resulting exotherm, or engine modifications to reduce the concentration of pollutants in the exhaust before the catalyst.

A number of researchers have applied oxidation catalytic converters to small two-stroke engines. Researchers at Graz University of Technology, ITRI in Taiwan and several other organizations have all published data on the application of catalytic converters in small two-stroke engines (Mooney et al., 1975; Engler et al., 1989; Burrahm et al., 1991; Laimbock and Landerl, 1990; Laimbock 1991; Hsien et al, 1992; Pfeifer et al., 1993; Gulati et al., 1993; Castagna et al., 1993). Some of these studies are discussed in detail later in this section.

As a result of this research, catalytic converters have been used on commercial production two-stroke motorcycles and mopeds in order to meet emissions standards in Taiwan, Switzerland, and Austria. Experience with these systems in consumer use has shown them to be acceptable, except that special heat shielding is necessary to protect the passengers from contact with the catalyst housing, which can have a skin temperature exceeding 500 °C. In Europe, catalytic converters have also been available since 1989 on production model chainsaws - with the primary intention to reduce inhalation of hydrocarbon emissions by the operators. These are presently an expensive option, found mostly on professional saws.

Graz University of Technology - The Graz researchers focused on reducing exhaust emissions from two-stroke moped, motorcycle and chainsaw engines by using catalytic converters, as well as by improving the thermodynamic characteristics of the engine, through changes in gas exchange and fuel handling systems, cylinder and piston geometry, and exhaust and cooling systems. Table 8 shows the effects of catalytic converters on emissions from production, lean-burn production, and advanced moped engines tested under the ECE-15 driving cycle. As the table shows, addition of a catalytic converter to the conventional moped reduced HC and CO emissions by 64 and 61%, respectively. The relatively low efficiency in this case was due to the rich air-fuel mixture used in the conventional moped, which limited the ability of the catalyst to oxidize the excess HC.

Comparing baseline emissions between the conventional moped and the lean-burn production moped, Table 8 shows that the lean-burn moped produced about 80% less CO and 18% less HC emissions, without the catalytic converter. The efficiency of the catalytic converter was also increased, due to the higher oxygen concentration in the exhaust. In this case, the CO and HC reductions were 75% and 89%, respectively. Durability testing on two production lean-burn Puch mopeds equipped with catalytic converters showed that the HC emissions increased by 42% while the CO emissions were reduced by 31% after 10,200 km or about 450 hours. At that point, the emission levels still met the Swiss standards. Since the overall air-fuel mixture was

Table 8: Emission data for production and advanced moped engines with and without catalyst tested under ECE-R47 driving cycle.

Engine Configuration		Emissions (g/km)		
		CO	HC	NOx
Production Puch	Without Catalyst	5.6	3.9	n/a
	With Catalyst	2.2	1.4	n/a
Production Puch Superm-axi, Lean-Burn	Without Catalyst	1.1	3.2	n/a
	With Catalyst	0.28	0.34	n/a
Advanced 1.2 hp	Without Catalyst	1.311	2.432	0.038
	With Catalyst	0.036	0.094	0.031
Advanced 2.7 hp	Without Catalyst	0.771	3.205	0.090
	With Catalyst	0.093	0.116	0.065
	With Dual Catalyst	0.022	0.037	0.067

rich, the catalyst oxidized HC to CO. The increase in HC and reduction in CO are consistent with a fairly rapid decline in catalyst efficiency with age. This would be expected, given the high operating temperature of the catalyst.

Table 8 also shows emission results for two advanced-technology moped engines of 1.2 and 2.7 HP. These engines were designed to operate near or lean of stoichiometric over almost the entire speed-load range. Addition of a catalytic converter to the 1.2 HP advanced moped engine reduced HC, CO and NOx emissions by 96, 97 and 18%, respectively. For the advanced 2.7 hp moped engine, the HC, CO, and NOx emissions were reduced by 96, 88 and 29% respectively when a catalytic converter was used. Data on catalyst temperatures were not provided in the Graz papers.

The Graz researchers also developed an advanced chainsaw engine equipped with a catalytic converter (Laimbock, 1991). In addition to the catalytic converter, this engine incorporated a new cylinder with four transfer ports, better cooling for the cylinder and cylinder head, and an optimized piston shape. Unlike the mopeds, this engine operated rich of stoichiometric. Engine maps showing emission results vs. speed and BMEP were presented in the Laimbock paper. For the chainsaw without catalyst, the engine map showed CO emissions ranging from 0.5 to 4.5%, HC emissions from 15,000 to 29,000 ppm, and NOx emissions from 30 to 400 ppm, depending on the load/speed conditions. For the chainsaw with catalyst, the CO, HC, and NOx emissions ranged from 0.5 to 4.7%, 7,000 to 17,000 ppm, and 3 to 300 ppm, respectively. These results translated into changes in CO emissions from about a 40% *increase* to an 80% reduction; 20 to 80% reduction in HC; and a 20 to 85% reduction for NOx. It should be noted that the NOx increased by 100% in a spot with high BMEP and air-fuel ratio slightly above stoichiometric.

For chainsaw engine emission development work, Graz University of Technology (G.U.T.) has developed a special emissions test cycle. This cycle is intended to simulate the main operation modes of chainsaws used by professional woodcutters, namely cutting and debranching operations. Laimbock (1993) reported that the typical emission results for standard production chainsaws, depending on the adjustment of the carburetor, were 3.7-5.9 g/min of HC emissions, 7.7-11.1 g/min of CO emissions, and 0.002-0.009 g/min of NOx emissions based on the GUT cycle. When the catalyst-equipped chainsaw was tested on GUT cycle, the emission results were 0.47 g/min for HC, 1.03 g/min for CO, and 0.028 g/min for NOx emissions. Comparing these results with the standard production chainsaws, average emission reductions for CO and HC were about 90%, while the NOx emissions slightly increased by 4%. Again, data on catalyst and tailpipe-out exhaust temperatures were not provided in the Graz papers.

It should be noted that in the work at Graz, catalytic converter efficiencies of 90% for HC and CO emissions were obtained mainly by the application of metal substrate technology along with lean air-fuel ratios.

<u>Industrial Technology Research Institute</u> - Researchers at ITRI have successfully retrofitted a catalytic converter to a 125 cc two-stroke motorcycle engine, and demonstrated both effective emissions control and durability (Hsien et al, 1992). The ITRI researchers evaluated the effects of catalyst composition and substrate, the cell density of the substrate, the converter size and installation location, and the use of secondary air injection on the catalytic effectiveness and engine performance. Their conclusions were as follows: 1) the use of a metal substrate is superior to the ceramic substrate of the same converter size in terms of conversion efficiency and engine performance, since the thin walls of the metal substrate result in a larger effective area and lower back pressure, 2) exhaust temperature profile, space availability, and the effects on engine exhaust tuning must be considered when installing the catalytic converter, 3) use of additional reduction catalyst Rh would improve the CO conversion efficiency in the rich air/fuel mixture environment typical of motorcycle two-stroke engines, 4) the cell density of the substrate should be less than 200 cpsi to minimize pressure loss and maintain engine power, 5) HC and CO conversion efficiencies increase significantly when secondary air is supplied, and 6) exhaust smoke opacity was also reduced with the use of the catalytic converter. This latter effect was due to the catalytic oxidation of the lubricating oil vapor in the catalytic converter. This effect has also been observed in other engines (Pfeifer et al., 1993).

In a more recent study, ITRI retrofitted a catalytic converter to a two-stroke scooter engine, together with fuel injection and skip-firing at idle. The emissions data were shown previously in Table 6. Adding the catalytic converter to the other emission control techniques improved the overall emissions control efficiency from 58.2% to 92.8% for HC, and from 56.8 to 97.6% for CO emissions. Efficiency improved only slightly with the use of secondary air. This is because the fuel-injected engine tested was able to operate with a lean mixture overall, so that sufficient oxygen was available in the catalytic converter even without the air injection.

<u>Other Catalyst Research on Two-Stroke Engines</u> - United Emission Catalyst (UEC) has investigated the use of catalysts in leaf blowers to determine the maximum reduction of HC and CO emissions possible using a catalyst size limited to that which will fit in a standard muffler

housing (Hobbs, 1995). A leaf blower engine was tested with and without catalyst at idle and WOT conditions. Emission results with a 64 cell catalyst showed 58% and 50% reduction in HC emissions at idle and WOT conditions, respectively. The CO emissions were reduced by 25% at idle and 49% at WOT conditions. However, the exhaust outlet temperature was increased from 95 to 145°C at idle, and 250 to 370 °C at WOT.

Pfeifer et al. (1993) also studied the effects of catalytic converters on exhaust HC, smoke and PM emissions for two-stroke motorcycle engines. The results showed that the catalytic converter not only reduced HC emissions, but PM emissions as well. Thus, the use of catalytic converters on two-stroke engines will significantly reduce PM emissions and smoke as well as HC and CO emissions.

Stihl Production Chainsaws Equipped with Catalytic Converters - Stihl is selling three models of chainsaws equipped with catalytic converters. The engine sizes of these chainsaws range from 49 to 77 cc. The average weight increase for these catalyst chainsaws ranges from 0.44 to 0.66 lbs (0.2-0.3 kg). Emission results obtained from Stihl for a 70 cc chainsaw with enleanment (air-fuel ratio of 16:1) and catalytic converter showed HC emissions reduced by 85% (from 85 to 10 g/bhp-hr), and CO emissions reduced by 45% (from 440 to 190 g/bhp-hr). However, even with heat shields and mixing ambient air with exhaust, Stihl reported that the maximum exhaust gas stream and skin temperatures were 530 and 300°C, respectively. These temperatures exceeded the US Forest Service's allowable temperature limits, which are 246°C for exhaust gas temperature and 289°C for skin temperature. In addition, these catalytic converters add about $100 to the price of the chainsaw. The added cost is said to be due to the costs for the catalytic element and its support, additional heat resistant material for the muffler, additional structural material for cooling and redirecting hot gases, and development and tooling costs. These latter probably account for the largest share of the increase, due to the very small volume of units over which they are spread.

5.6 Lubricating Oil Technologies

Lubricating oil is the major source of PM emissions from two-stroke engines. Since the crankcase of a two-stroke engine is used for pumping air or a fuel-air mixture to the combustion chamber, it cannot also act as a lubricant-oil reservoir. Instead, a fine mist of oil is injected into the incoming air stream. As this stream passes through the crankcase, lubrication is provided for cylinder walls, and crankshaft and connecting-rod bearings. Ball or roller bearings are typically used instead of a four-stroke engine's plain bearings. The oil mist continues to the combustion chamber, where some of it is trapped and burned. Oil that is not trapped in the combustion chamber, or which survives the combustion in the chamber, recondenses in the exhaust plume to create the blue or white smoke that is the distinguishing characteristic of the two-stroke engine. Any phosphorus or other deposit-forming additives in the two-stroke oil can also be expected to poison the catalytic converter, reducing its efficiency. Thus, two-stroke oils for catalyst-equipped motorcycles or utility equipment will need to be formulated without these compounds.

Table 9: Comparison of particulate emissions from a two-stroke motorcycle engine lubricated with mineral oil and a low-smoke PIB oil.

Particulate Emissions (g/600 liter Exhaust Gas)			
Fuel Oil Ratio	**20 : 1**	**40 : 1**	**60 : 1**
Conventional Mineral Oil			
Solid Component	0.009	0.005	0.006
Oil Component	0.304	0.247	0.204
Total	0.313	0.252	0.210
Low-Smoke Oil With PIB			
Solid Component	0.008	0.005	0.004
Oil Component	0.228	0.120	0.061
Total	0.236	0.125	0.065

Source: Sugiura et al (1977).

Lubrication system

Three approaches are commonly used to supply lubricating oil to two-stroke engines: pre-mixing with the fuel when it is added to the tank; line-mixing in which the oil is metered into the fuel between the fuel tank and the engine; and oil injection, in which the lubricating oil is metered directly into the intake manifold or other points using a pump controlled by engine speed and/or throttle setting. The last two approaches are common in motorcycle engines, as they have the ability to control the flow rate of the lubricating oil and provide more reliable lubrication. For cost reasons, however, nearly all hand-held equipment engines premix the oil with the fuel. Most chainsaws do have an automatic lube oil feeder, but this is for the chain lubricant, not the engine oil.

The injection-type lubricating oil metering system provides the best control of oil metering. Orbital has designed an electronic lubrication system for their OCP two-stroke engines to reduce the amount of oil required by the engine. Several models of Yamaha two-stroke motorcycles marketed in Asia have also used an electronic lubricating oil metering system to alter the lubricating oil flow to the carburetor according to the engine load demand. The Yamaha Computer-Controlled Lubrication System (YCLS) supplies the required amount of lubricating oil to the engine according to the engine speed, using an electronic control unit and three-way control valve. In a fuel-injected two-stroke engine, this function could be handled by the same electronic control unit as the fuel injection system.

"Low-smoke" oil

Conventional two-stroke lubricating oils are based on long-chain paraffin or naphthene molecules that break down slowly under combustion conditions, and are thus resistant to combustion. The use of synthetic long-chain polyolefin materials instead of naphthenes and paraffins can significantly reduce smoke opacity and particulate emissions from two-stroke engines. Because of the periodic occurrence of double carbon bonds in the polyolefin chain, these chains break down much more rapidly and thus burn more completely in the two-stroke engine. Studies by Souillard (Souillard et al., 1971), Sugiura (Sugiura et al., 1977), Kagaya (Kagaya et al., 1988), and Eberan-Eberhorst (Eberan-Eberhorst et al, 1979) have provided ample evidence that substitution of polyisobutylene (PIB) for bright stock or other heavy lube-oil fractions in two-stroke lubricating oils can reduce engine smoke levels and particulate emissions. An experiment performed by Broun of Lubrizol further demonstrated the decrease in smoke levels using such materials (Broun et al, 1989). Results of lubricity tests by independent laboratories using lubricating oil with PIB and bright stock showed that the lubricity performance for both lubricating oils was essentially the same.

In addition to smoke levels, Sugiura et al.'s study also investigated the effect of PIB on particulate emissions (Sugiura et al., 1977). A comparison of the particulate emissions with a conventional oil and oil with PIB is shown in Table 9. As this table shows, substantial reductions in particulate emissions were achieved using oil with PIB, ranging from 25 to 70% depending on the fuel/oil ratio. It also shows that the leaner the fuel/oil ratios, the lesser the particulate emissions, especially for the oil with PIB. Thus, higher particulate emission reductions were observed with the oil with PIB at leaner fuel/oil ratio as compared to conventional oil.

While research is still under way to formulate better lubricating oils for two-stroke engines, a "low smoke" polyisobutene based lubricating oil is being required to be used for two-stroke mopeds and motorcycles in some of the countries of Southeast Asia, including Thailand. The current Japanese standard and a proposed International Standards Organization (ISO) standard for two-stroke oil include a special category of low-smoke oils.

Although probably helpful, the use of low-smoke lubricating oils alone will not solve the particulate problem for two-strokes. The oil contained in the 20-30% of the fresh charge that short-circuits the cylinder will be unaffected by combustion. In addition, there is a possibility that the combustion of the polyisobutene lube stock may increase emissions of toxic air contaminants, especially 1,3 butadiene. Further laboratory research is needed to assess the real effects of these oils on particulate and other emissions. In testing to be conducted in Thailand in March, 1996, EF&EE will attempt to measure the benefits of these oils in reducing PM emissions from two-stroke motorcycles.

5.7 Conversion to Four-Stroke Engines

As discussed previously, four-stroke engines inherently produce less HC, CO, and particulate emissions than two-stroke engines. Therefore, one way to reduce emissions from utility equipment using two-stroke engines is to convert the equipment to use four-stroke engines if the applications of the equipment permit. Nearly all of the two-stroke engines used in the non-handheld equipment were replaced by four-stroke designs after the ARB Phase I emission regulations took effect. However, in order to use four-stroke engines in handheld equipment, the engine has to provide the inherent advantages of a two-stroke engine. These include high power to weight ratio, compactness, and multiposition operational capability. Since four-stroke engines have more mechanical parts, the cost to produce them is also higher. In order to be competitive, the cost of the four-stroke engine would need to be comparable to or less than that of the advanced two-stroke engines that could also would meet the Tier II standards.

Ryobi Four-Stroke Engine

Recently, Ryobi certified a string trimmer powered by a four-stroke engine to meet the Tier I emission standards. This innovative 26 cc four-stroke engine, which uses overhead valve and exhaust gas recirculation (EGR) technologies, was the result of many years of research and development work in Ryobi. The engine is rated at 0.75 to 1 hp, and has engine life of about 100 to 200 hours. Ryobi also reported that it can build handheld four-stroke engines up to 3 horsepower.

Emission Levels - The Ryobi four-stroke engine is the first and only handheld four-stroke gasoline engine certified with the ARB to meet the Tier I emission standard. The certified emission data are 368 g/bhp-hr for CO, 28 g/bhp-hr for HC, and 3.0 g/bhp-hr for NOx emissions. The HC and CO values were determined with the carburetor set in the rich/rich position, and the NOx value was determine with the carburetor set in the lean/lean position. The HC and NOx emissions were below the Tier II limits, but the CO emissions exceeded the Tier II standard. Emission data supplied by Ryobi indicated that all three emissions were below the Tier II emission standards with the carburetor set in the lean/lean position.

While the current emission levels for this engine do not meet the Tier II emission standards, Ryobi is continuing refine the engine by limiting carburetor adjustments, as well as using different cam designs and valve timing to control NOx emissions without EGR. The reason for investigating other means to control NOx emissions is that it has been reported that the EGR passage in the engine might plug when the carburetor is set at the rich/rich position. While CO and NOx emissions were marginally below the Tier II emission standards, the emission results for some of these approaches met the Tier II emissions standards.

Ryobi also tested the four-stroke engine with a catalytic converter. The results showed that HC, CO and NOx emissions were reduced and met the Tier II emission standards. A substantial reduction in NOx emissions was observed (from 3.1 to 0.5 g/bhp-hr). However, as with other research on catalytic converters for two-stroke engines, the skin and exhaust temperatures were increased and exceeded the US Forest Service limits. All these results indicate that more testing is necessary with different emission control approaches to reduce the CO emissions while maintaining the NOx emissions or vise versa.

Cost and Weight - Compared to a similar 31 cc two-stroke model that Ryobi offers, the four-stroke string trimmer costs about 50 to 80% more at retail, depending on the design features, and it weighs about 2 to 3 lbs more based on the same specifications. Ryobi reported that the four-stroke engine alone only weighs about 1.6 lbs more than the comparable two-stroke model. EF&EE weighed individual components of both the four-stroke and two-stroke engines, and found that the four-stroke engine is about 2 lbs heavier. The major weight differences came from the crankcase, cylinder head and clutch assembly of the four-stroke engine, which each contributed about 0.6 lbs in weight increase. Ryobi also reported that the differential weight can be reduced to less than 0.6 lbs by retooling some injection molded and die cast components.

Engine Features - EF&EE also compared the number of parts in the two-stroke and four-stroke engines. The Ryobi four-stroke engine had only about 40 more parts. Typical four-stroke engines with overhead valves, such as those used in motorcycles, often have more than 50 more parts than a two-stroke. Ryobi has incorporated many features to reduce the weight and parts in this engine, which in turn make it more compact and less expensive to produce. The innovative design of the cylinder head assembly, which includes the miniature overhead valve train and EGR passage, allow it to tolerate high speeds and loads. Many of the valve train parts, including the intake and exhaust valves and followers, are interchangeable to reduce manufacturing cost. A simple gear-lobe-follower assembly is used to control the valve timing instead of the typical two-lobe camshaft used in many engines with overhead valve configuration.

A small passage is drilled between the intake and exhaust ports to provide EGR. An accelerator pump is also used in this engine to keep the carburetor from going too lean during acceleration. The engine uses the splash lubrication method to lubricate the crankshaft bearing, piston/cylinder and valve train assembly. While Ryobi tried to design the current string trimmer to have multiposition operational capability, field tests showed that the engine began smoking and oil was dripping from the air filter when the trimmer was operated for a few minutes with the exhaust side down. However, the string trimmers do come with a split boom design, which allows the operator to adjust the front part of the trimmer to perform edging while still keeping the engine upright.

The engine uses a solid state ignition system with retard for easy starting. It also has an electronic speed limiter incorporated in the ignition module to interrupt the spark to cause the engine to miss if the engine speed becomes too high.

Current Status - With the intention of utilizing the four-stroke engine in other handheld equipment, Ryobi is continuing to refine and finalize the technology to meet the Tier II emission standards, as well as to obtain multiposition operating capability. Ryobi has also indicated that it is ready and willing to license its four-stroke engine design to other manufacturers. It has been reported that Ryobi is involved in licensing discussions with several manufacturers, and at least one company is buying Ryobi four-stroke trimmers to be sold under its own brand name.

With Ryobi's demonstration of the feasibility of using small four-stroke engines on string trimmers, many handheld engine and equipment manufacturers are believed to be considering four-stroke engine technology as one of their research and development alternatives to produce low emission handheld equipment, at least for those applications that do not require total multiposition operational capability. These applications include string trimmers and blowers, but not chainsaws. Small four-stroke engine technology is well developed and understood in motorcycle and non-handheld equipment applications. Thus, it will not be surprising to see other four-stroke engines emerging for the handheld equipment if that is one of the viable technologies to meet the emission regulations.

6. TECHNOLOGICAL APPROACHES AND INCREMENTAL COST OF MEETING TIER II EMISSION STANDARDS

Based on the technology review in Chapter 5, this chapter describes several promising technological approaches for meeting ARB's Tier II emission standards for handheld equipment engines. The increment cost to manufacturers and consumers, and the cost-effectiveness of the standards are also estimated, assuming that these technologies were implemented on a mass production basis.

6.1 Technological Approaches

For a two-stroke engine to meet the Tier II emission standards, EF&EE believes that an oxidation catalytic converter will be required. However, the use of catalyst technology along is not sufficient to meet both the emission limits and the U.S. Forest Service limits on exhaust temperature. If a catalytic converter with a high efficiency were used, the exothermic energy released by the oxidation of HC and CO would be very high. The resulting catalyst temperature would exceed the thermal limits of the catalytic converter, as well as exceeding the USFS limits on exhaust and skin temperatures. If a catalytic converter with a low efficiency were used, the emission reductions would not be sufficient to meet the Tier II standards. Thus, the key requirement in each of these approaches is to reduce the engine-out HC and CO emission levels to the point that a catalytic converter can survive in the exhaust without overheating, and to achieve an overall lean or stoichiometric air-fuel ratio in the exhaust to maximize catalytic converter efficiency. It is then possible to rely on the catalytic converter to bring the remaining HC and CO to levels well below the applicable standards.

As a first step toward reducing engine-out HC and CO emissions, handheld equipment engines that already meet ARB Tier I emission standards should be used as the starting point. Compared to previous, uncontrolled handheld equipment engines, Tier I engines are generally calibrated for a leaner air-fuel ratio, and their cylinder cooling systems are designed accordingly. In addition, Tier I engines are generally equipped with somewhat more precise air-fuel ratio control systems to reduce the variation in air-fuel ratio, emissions, and performance.

EF&EE performed an analysis to determine the catalytic converter temperature in a two-stroke engine as a function of pollutant concentration. The analysis is documented in our proposal for this study (EF&EE, 1993). The engine parameters and emissions data used in the analysis were based on a chainsaw engine, and are listed in Table 10. Emission concentration data for this engine - operating at full power - were provided by a chainsaw manufacturer. As the table shows, the chainsaw used in the analysis was well within ARB Tier I emission standards. To assess the catalyst temperatures that would result from fitting a catalyst to this chainsaw, EF&EE developed a simplified thermodynamic model of the catalytic oxidation process. A maximum air injection rate before the catalyst of about 30 CFM was found to be possible without reducing the exhaust temperature below 300°C, which was considered essential in order to provide catalyst light-off. With this rate of air injection, and the pollutant concentrations for the baseline engine, the maximum temperature in the catalytic converter was calculated to be 1600 °C. This is well above the maximum allowable catalyst temperature - in fact, it exceeds the melting points of both ceramic and metal catalyst supports. The results of this analysis thus confirmed that pollutant concentrations in the exhaust must be reduced significantly from those needed to meet the Tier I standards before a catalytic converter would become feasible. It should be noted that the air/fuel ratio in this case was not especially rich by small equipment standards, with an exhaust CO concentration of about 4%.

Table 10: Chainsaw engine parameters and emissions data.

Engine Parameters	
Displacement (cc)	50
Horsepower	3.55
Rated Speed (rpm)	11,000
Emissions Data	
CO (g/bhp-hr)	195
CO (% Vol)	4
HC (g/bhp-hr)	104
NOx (g/bhp-hr)	1.1

A wide variety of emission control measures and design features could be used to achieve the further reduction in engine-out HC and CO emissions needed to allow the catalytic converter to survive in the exhaust. Of these, the most practical, in EF&EE's view, are direct in-cylinder fuel injection, indirect fuel injection into the transfer port, and stratified scavenging by filling the transfer ports with air. Meeting the PM standards poses somewhat different challenges from meeting the HC and CO standards. The catalytic converter can be expected to reduce PM emissions by 60-80%, by oxidizing much of the lubricating oil vapor in the exhaust. PM emissions can be further reduced, if necessary, by the use of low-smoke lubricating oil containing PIB, and by measures to reduce the amount of lubricating oil lost in the exhaust.

All the approaches discussed above are based on two-stroke engines. An alternative approach would be to convert from two-stroke engines to four-stroke engines similar to those used in Ryobi string trimmers. Each of these alternatives is examined in more detail in the following sections.

Direct In-Cylinder Fuel Injection

The use of direct in-cylinder fuel injection has the advantage that scavenging losses can be eliminated completely, since fuel injection can be delayed until the exhaust port is closed or nearly closed. Light-load HC emissions can also be reduced by skip-firing. At the same time, eliminating the need to compromise between adequate scavenging and high HC emissions and fuel consumption should make it possible to increase scavenging airflow through the cylinder, thus improving scavenging efficiency, reducing cylinder temperature somewhat, and permitting equal or better power output at a leaner air-fuel ratio. The direct fuel injection system could be mechanically or electronically operated. The Stihl mechanical fuel injection system (Figure 13) and the Orbital SEFIS system (Figure 11) are excellent examples of well-engineered direct fuel injection systems for small two-stroke engines. Both these systems use crankcase pressure fluctuations to pressurize the fuel. A catalytic converter with moderate efficiency would be needed to reduce the PM emissions, as well as to further reduce the gaseous emissions.

The main disadvantage to this approach is that the fuel injection system would add significantly to the costs of production. Another disadvantage is that crankcase lubrication by adding oil to the fuel would no longer be possible, making it necessary to develop a separate lubrication system. However, the use of a separate lubrication system should make it possible to reduce total lube oil use, and therefore to cut down on engine-out PM emissions.

The main design issues to be addressed in the direct fuel injection system are:

1. achieving sufficiently rapid vaporization and mixing of air and fuel in the cylinder at WOT;
2. providing electric power to operate the fuel injector, especially when starting the engine (for electronic controlled fuel injection system);
3. pressurizing the fuel (and air, for air-assisted injection) prior to injection;
4. electronic control of fuel injection and ignition timing (for electronic controlled fuel injection system);
5. lubricating the piston and crankshaft bearings, while minimizing oil carryover into the exhaust to control PM emissions.

Indirect Fuel Injection

The indirect fuel injection approach would resemble the direct fuel injection system outlined above, except that fuel would not be injected directly into the combustion chamber, but into the transfer port instead. This would allow more time for the fuel to evaporate, and thereby eliminate the need for high pressure or air-assistance. By appropriate timing of the fuel injection, it should be possible to reduce scavenging losses very substantially. However, these losses would still be greater than with direct injection. This approach would also allow skip-firing to reduce HC emissions at idle and light load if an electronic fuel injection system is used. A catalytic converter with high efficiency would be needed to reduce the PM emissions and to further reduce gaseous emissions. The main disadvantages would be the same as for the direct-injection system - cost, and the need for a separate lubrication system.

The main design issues to be addressed in this approach would be the following:

1. optimizing injection timing and port configuration to minimize HC carryover into the exhaust at WOT, while still assuring a reasonably homogeneous charge and a combustible mixture at the spark plug;

2. providing electric power to operate the fuel injector, especially when starting the engine (for electronic fuel injection system);
3. pressurizing the fuel prior to injection;
4. electronic control of fuel injection and ignition timing (for electronic fuel injection system);
5. lubricating the piston and crankshaft bearings, while minimizing oil carryover into the exhaust to control PM emissions.

This approach involves significantly less technical risk than the high pressure or air-assisted direct injection system, since the fuel injection system requirements are not too different from those of present automotive technology. However, the emissions performance is not likely to be as good (though it should still meet the Tier II standards).

Stratified Scavenging

The stratified scavenging approach would involve changes in the air system to prefill the transfer ports with air instead of air-fuel mixture. This could be expected to reduce full-power HC emissions by at least 30%, and possibly as much as 50% with some effort spent on optimizing the scavenging flows. At the same time, an exhaust control valve - actuated by the throttle trigger - would be used to reduce misfire and HC emissions at idle. It should be noted that optimizing emissions at idle and full-power operation at a single speed, as required by the two-mode test procedure, would be fairly straightforward. "Real world" emissions under transient conditions and varying speeds would likely be considerably higher, however. This would pose serious concerns about possible catalyst damage due to overheating under real-world conditions, as well as excess emissions due to "off-cycle" operation.

The main design issues to be addressed with this approach are the following:

1. Design of separate intake systems to prefill transfer ports with air;
2. Optimizing transfer port configuration to minimize scavenging losses while still assuring a reasonably homogeneous charge and combustible mixture at the spark plug;
3. Exhaust control valve or other means to improve idle performance;
4. Means to reduce lubricating oil carryover into the exhaust.

Conversion to Four-Stroke Engines

Converting to four-stroke engines seems to be a possibility for handheld equipment types that do not require multiposition operating capability. These include string trimmers and blowers, but not chainsaws. As discussed previously, Ryobi has demonstrated the use of four-stroke engines in string trimmers. The main disadvantages would be the costs, the need for a crankcase lubrication system, and the loss of multiposition operating capability.

6.2 Manufacturing and Consumer Costs

To bring some order and reproducibility to cost estimates of emission control systems, EPA has developed a standard retail price equivalent (RPE) technique (Jack Faucett Associates, 1985). EPA's RPE methods were first outlined by Lindgren (Lindgren ,1978) in a study done for EPA in 1978, and refined by Putnam, Hayes, and Bartlett (PHB, 1984). The basic equation for the retail price equivalent (RPE) of a given vehicle or engine modification is given by:

$$RPE = ((SP + AL + AO) * MM + RD + TE + WC) * DM$$

where:

RPE is the retail price equivalent;

SP is the supplier price charged to the auto assembler for the components and subassemblies involved;

AL is the direct cost of assembly labor for installing the components;

AO is the manufacturer's assembly overhead cost per unit;

MM is the manufacturer's markup percentage, to account for corporate overhead and profit;

RD is the manufacturer's research and development cost per unit;

TE is the manufacturer's tooling cost per unit

WC is the manufacturer's added warranty cost, per unit; and

DM is the dealer's markup percentage.

In order to obtain the information needed to calculate the RPE, EF&EE solicited small engine manufacturers and suppliers to provide data and estimates for each technology approach (see Appendix A). No useful responses to this solicitation were received. EF&EE staff also contacted industry associations, who provided some general indications of typical markups by manufacturers and retailers, and researched the available literature for relevant information. This included the results of the most recent Census of Manufactures for lawn and garden equipment manufacturers. No similar data were available at the retail level.

To determine typical dealer markup percentages, EF&EE contacted the North American Equipment Dealers' Association. According to an association spokesperson, the typical dealer markup for handheld equipment ranges from 16 to 30 percent, with the lower range being typical of "consumer" equipment, and the upper range being typical for "professional" grade units. The typical markup has declined substantially in recent years, due to competition from warehouse stores and other large-volume retailers. For our calculations, we assumed a dealer markup at the midpoint of the range cited, or 23%.

The Portable Power Equipment Manufacturers' Association was contacted for information about manufacturer markups. According to the PPEMA spokesperson contacted, typical manufacturer's markup is from 5 to 10% over costs. EF&EE also analyzed the data for the lawn and garden equipment industry in the 1992 Census of Manufactures. These data showed that, for the lawn and garden equipment industry as a whole, variable production costs were equal to 70% of the total value of shipments in 1992, while fixed production costs and overheads accounted for 8%. Of the 70% variable production costs, production wages and fringe amounted to only 10%, while 60% were due to material costs. The 22% of value of shipments not accounted for by variable costs, fixed production costs, or overheads was assumed to constitute returns to capital - interest on debt and return to stockholders. If one assumes that the costs of capital were included in base costs, these figures are consistent with the PPEMA spokesman's estimate of 5 to 10% markup. The incremental capital costs of meeting the emission standards are explicitly factored into our estimates, so they are not included in the assumed markup to avoid double counting. The assumed markup was taken as the midpoint of the range cited by the PPEMA spokesman, or 7.5%.

Developing handheld equipment engines that meet the ARB Tier II emission standards will involve a significant research and development effort. EF&EE estimated the research and development costs based on direct labor costs, emission testing costs and other R&D costs. Twelve handheld equipment engine manufacturers certify their products with ARB. We assumed that the seven largest manufacturers would devote major R&D efforts to develop engines that meet the Tier II emission standards. The smaller manufacturers would be more likely to depend on suppliers and/or licensing available technologies, and would assign only a few of their own staff to this effort. Assuming that each of the major manufacturers assigned an average of four full-time equivalent R&D staff, this would amount to 28 engineers working full time on the R&D component. Another 12 or so would be employed by the minor manufacturers and technology suppliers. Thus, the total R&D effort would amount to about 40 engineer-years per year. For typical engineering salaries and overhead rates, the cost of an engineer working full-time for a year (including salary, benefits, physical and administrative overhead, and other costs) is assumed to be about $100,000. With this loaded engineer cost, the total cost of R&D staff time alone for developing the Tier II engines would be about 4 million dollars.

The costs of test engines, emission testing, special materials, travel, and other similar expenses would add substantially to the R&D cost. For instance, EF&EE estimates that the development effort would require about 500 emission tests per year per manufacturer, which is about two tests per day per manufacturer. With seven major manufacturers, the total emission tests would amount to 3,500 tests per year. Assuming another 500 emission tests that would be performed by minor manufacturers and suppliers, the total emission tests would be 4,000. Assuming $250 per test, the emission tests would cost about one million dollars. Assuming test engines, travel, other R&D and test materials and so forth would cost another million dollars, the total R&D costs would likely be about 6 million dollars per year.

This R&D effort is under way now in response to Tier II emission standards, and will likely continue through 1997. By 1997, the problem will need to have been solved in order to proceed with 1997 production. Thus, assuming that successful development takes place in the time for the 1997 production, the total cost of the R&D effort should be about three years of 6 million dollars or 18 million dollars. With interest at 10% per year, and amortizing the costs over 10 years, the amortized R&D costs for developing Tier II engines would be about 3 million dollars per year. Based on ARB's handheld equipment inventory document (ARB, 1990), the average annual engine sales for handheld equipment in California are about 500,000 units. Thus, the R&D cost per engine is about six dollars.

Designing, developing and producing new engines would also require tooling costs. Since almost all major engine manufacturers should include some form of tooling costs throughout the development of new products regardless of emission regulation, EF&EE estimates that each major manufacturer would spend about half a million dollars on tooling costs as a result of emission regulations. Thus, this would amount to about 3.5 million dollars for seven manufacturers. Amortizing the 3.5 million dollars tooling costs at 10% interest over ten years yields about one dollar per engine for an average annual engine sale of 500,000 units.

The incremental direct cost of assembly labor and manufacturer's assembly overhead cost for installing emission control components are assumed to be negligible compared to the cost of the components. Since there is no in-use compliance requirement, the manufacturer's added warranty cost for the added components is omitted in the analysis. The estimated costs of added materials for each technology approach are also estimated based on information provided by the suppliers and manufacturers. These estimates along with the R&D costs, emission testing costs, tooling costs, as well as the total RPE, are presented in the following sections for each technological approach.

Direct Fuel Injection System

The incremental component costs of a handheld equipment engine equipped with a catalytic converter and a direct injection system would be very significant. Table 11 shows our estimate of the incremental cost components for such a system to the manufacturer, assuming mass production (about 40,000 to 50,000 units per year). The main cost elements are the air-fuel injector and the electronic control unit. The catalytic converter cost assumes a ceramic monolith substrate (a metal substrate would cost about $20). A pulse alternator would be required to supply the electric power to drive the fuel injector. EF&EE assumes that the fuel pump and lube oil metering pumps would both be low-cost units driven by crankcase pressure fluctuation.

The total incremental component cost to the manufacturer for this system is estimated at $44. While expensive to manufacture, there is relatively little risk that a system of this type would not work, or that it would fail to meet ARB Tier II emission standards. The two main technical issues to be resolved would be developing a fuel injector fast enough to operate at 8 to 12,000

RPM, and providing adequate lubrication through a separate lube-oil system. The latter may require some experimentation, but is not fundamentally difficult. Therefore, except for the actual fuel injector, development of a prototype system of this sort would be relatively straightforward.

Table 11 also shows the total RPE for this system based on the RPE methodology presented earlier. We assumed that the changes in components would not change assembly labor requirements significantly. The calculation also assumed no added warranty cost, since the Tier II standards do not impose recall liability for in-use performance. With these estimates and assumptions, the total incremental RPE to the consumer for a piece of handheld equipment with a catalytic converter and an electronic direct fuel injection system is about $66.

Table 11: Estimates of the retail price equivalent for an electronic air-assisted, direct injection handheld engine with catalytic converter in large volume.

Electronic Direct Fuel Injection System	
Hardware	Cost
Fuel/Air Injector	$15
Fuel Pump & Regulator	5
Air Pump	4
ECU	10
Pulse Alternator	2
Separate Lube System	5
Catalytic Converter	5
Double-wall Exhaust	3
Eliminate Carburetor	-5
Total Hardware	$44
Mfr's Markup @ 7.5%	3
Research & Development	6
Tooling	1
Total Cost to Dealer	$54
Dealer's Markup @ 23%	12
Retail Price Equivalent	**$66**

EF&EE also estimated the cost of a direct fuel injection system using mechanical means instead of electronics, such as the one used in Stihl's fuel-injected chainsaw. The major drawback of using a mechanical fuel injection system will be the loss of skip injection capability, which is essential for light load conditions. The estimates of hardware cost and incremental RPE are shown in Table 12. The cost for the fuel injector for this mechanical fuel injection system is assumed to be about $3 less than the air-assisted one, as it is less complex. However, the fuel pump and regulator assembly are assumed to be twice as much as the electronic version, as it would include many more mechanical elements to control and regulate the fuel. Obviously, the air pump, ECU and pulse generator are assumed to be unnecessary for this system. An estimate for an oil pump is included in this design. With these changes, the total incremental retail price to consumer for this design is $54.

Indirect Fuel Injection System

Table 13 shows our cost and RPE estimates for the indirect fuel injection system. The costs are similar to those for the direct injection system, except that the fuel injector is estimated to be somewhat less expensive, and there would be no need for a separate air pump driven by the crankcase pressure fluctuations. The estimated hardware costs for this system are $37, and the total incremental RPE to the consumer for this design is $58, which is slightly more than the mechanical direct fuel injection system. EF&EE also estimated the hardware costs and RPE for a mechanical indirect fuel injection system. These estimates are shown in Table 14. The hardware costs are essentially the same as the mechanical direct fuel injection system except the fuel pump and regulator. Since lower pressure fuel would be required for indirect fuel injection system, the fuel pump and regulator costs are assumed to be $4 less than the direct fuel injection system. With this estimate, the hardware costs and incremental RPE for the consumer for this design are $30 and $48, respectively - the cheapest among the fuel injection designs.

Table 12: Estimates of the retail price equivalent for a mechanical direct injection h andheld engine with catalytic converter in large volume.

Mechanical Direct Fuel Injection System	
Hardware	Cost
Fuel Injector	$12
Fuel Pump & Regulator	10
Oil Pump	4
Separate Lube System	5
Catalytic Converter	5
Double-wall Exhaust	3
Eliminate Carburetor	-5
Total Hardware	$34
Mfr's Markup @ 7.5%	3
Research & Development	6
Tooling	1
Total Cost to Dealer	$44
Dealer's Markup @ 23%	10
Retail Price Equivalent	**$54**

Stratified Scavenging

Since the use of fuel injection system substantially increases the material costs, EF&EE also estimated the hardware costs and RPE for a less complex emission control technology, namely stratified scavenging, to control scavenging losses. Although it is not certain that this approach would reduce emissions enough to meet the Tier II standards, it was considered interesting to determine the incremental cost for this approach. Table 15 shows our estimates for the hardware costs and the RPE for a system of this type. The hardware costs turned out to be $22. As would be expected, the hardware costs are significantly lower than the fuel-injected technology options. However, this alternative would entail fairly significant technical risks as extensive optimization efforts might be required to achieve a sufficient reduction in engine-out HC emissions. Thus, although hardware costs would be lower, development costs would likely be higher than with the fuel-injected systems as reflected in the RPE calculation. With this approach, the total incremental RPE is $39. Although the incremental RPE is lower than for the fuel injection options, engines using the stratified scavenging approach would entail greater risks of failing to meet the Tier II standards.

Converting to Four-Stroke Engine

Converting from two-stroke engines to four-stroke engines would involve additional costs for material and parts. EF&EE has estimated added material costs for motorcycle engines in Asia to be around $30 based on the retail prices for the engine parts (Chan and Weaver, 1994).

However, the manufacturer's and dealer's markups are different for Asian motorcycle and the U.S. handheld equipment manufacturers. In a letter to the EPA (Suchdev, 1995), Ryobi claims that the cost for added materials and parts for converting two-stroke engines to four-stroke engine will be about $10. This estimate is about $20 less than the EF&EE estimate for motorcycle engines. For this study, EF&EE assumed that the incremental material and part costs for converting two-stroke design to four-stroke design for handheld equipment is about $20. EF&EE also assumed that a catalytic converter would be required to meet the Tier II standards. The hardware costs, as well as the RPE for converting to four-stroke engine are shown in Table 16. Adding the costs for catalytic converter and double-wall exhaust system, the hardware costs for converting to four-stroke design amount to be $28, and the incremental RPE is $46. Although the incremental RPE is higher than the stratified scavenging approach, converting from two-stroke engine design to four-stroke design involves substantially less risks to meet the Tier II emission standards - at least for those applications that do not require total multiposition operational capability.

Table 13: Estimates of the retail price equivalent for an electronic indirect injection handheld engine with catalytic converter in large volume.

Indirect Fuel Injection System	
Hardware	Cost
Fuel Injector	$12
Fuel Pump & Regulator	5
ECU	10
Pulse Alternator	2
Separate Lube System	5
Catalytic Converter	5
Double-wall Exhaust	3
Eliminate Carburetor	-5
Total Hardware	$37
Mfr's Markup @ 7.5%	3
Research & Development	6
Tooling	1
Total Cost to Dealer	$47
Dealer's Markup @ 23%	11
Retail Price Equivalent	**$58**

6.3 Cost-Effectiveness Analysis

Handheld equipment engines that meet the proposed Tier II emission standards are expected to decrease fuel consumption significantly. The amount of savings on the fuel consumption depends on the intensity of utilization of the equipment. In ARB's nonroad equipment inventory study (ARB, 1990), the average lifespans for commercial and residential handheld equipment were estimated to be 2.2 and 5.0 years, respectively. The average annual usages were estimated to be 261 and 9.5 hours for commercial and residential handheld equipment, respectively. Using the estimates in the ARB study, an average fuel consumption of 1.35 lbs/bhp-hr was also assumed in the analysis. Average values of 3 and 1.5 horsepower were assumed for the commercial and residential handheld equipment, respectively. Assuming that all technological approaches would provide at least 30% reduction in fuel consumption, EF&EE has determined the value of the fuel savings for the commercial and residential handheld equipment to be about $151 and $6 per year, respectively.

EF&EE also calculated the cost-effectiveness of the technology approaches. Two ranges of cost-effectiveness were calculated based on the lowest and highest incremental RPE ($39 and $66), the projected emission reductions, and fuel savings. The projected emission reductions were calculated from the differences between the average emission levels from Tier I engines and the Tier II limits. The results of the cost-effectiveness analysis for the commercial and residential handheld equipment are presented in Table 17 and Table 18, respectively.

Table 14: Estimates of the retail price equivalent for a mechanical indirect injection handheld engine with catalytic converter in large volume.

Mechanical Direct Fuel Injection System	
Hardware	Cost
Fuel Injector	$12
Fuel Pump & Regulator	6
Oil Pump	4
Separate Lube System	5
Catalytic Converter	5
Double-wall Exhaust	3
Eliminate Carburetor	-5
Total Hardware	$30
Mfr's Markup @ 7.5%	2
Research & Development	6
Tooling	1
Total Cost to Dealer	$39
Dealer's Markup @ 23%	9
Retail Price Equivalent	**$48**

For commercial equipment, the cost-effectiveness calculations show that, even without considering the fuel saving, the costs per ton of VOC emissions eliminated would be far lower than the costs per ton for most other available emission control strategies. With the fuel savings counted in, the net costs are negative (i.e. the is a lifecycle cost saving), since the saving on fuel costs more than outweighs the higher purchase price.

The cost-effectiveness for residential handheld equipment is shown in Table 18. Without considering the fuel savings, although the costs per ton are substantially higher than those for the commercial handheld equipment, the cost-effectiveness numbers are still within the acceptable ranges. In addition, it should be considered that many residential users of hand-held equipment can substitute cord-electric units, which are significantly less costly even than present engine-powered units. Thus, it is likely that only the heavier residential users of handheld equipment would choose to purchase emission-controlled, engine-powered equipment rather than cord-electric or battery-electric units.

Table 15: Estimates of the retail price equivalent for a handheld engine with stratified scavenging technology and catalytic converter in large volume.

Stratified Scavenging	
Hardware	Cost
Reed Valves	$4
Added machining	5
Separate Throttle	1
Exhaust Control Valve	4
Catalytic Converter	5
Double-wall Exhaust	3
Total Hardware	$22
Mfr's Markup @ 7.5%	2
Research & Development	7
Tooling	1
Total Cost to Dealer	$32
Dealer's Markup @ 23%	7
Retail Price Equivalent	**$39**

Table 16: Estimates of the retail price equivalent for a four-stroke handheld engine with a catalytic converter in large volume.

Converting to Four-Stroke Engine	
Hardware	Cost
Added Part & Manufacturing Cost	$20
Catalytic Converter	5
Double-wall Exhaust	3
Total Hardware	$28
Mfr's Markup @ 7.5%	2
Research & Development	6
Tooling	1
Total Cost to Dealer	$37
Dealer's Markup @ 23%	9
Retail Price Equivalent	**$46**

Table 17: Cost-effectiveness of ARB Tier II emission standards for commercial handheld equipment.

Cost Effectiveness Analysis: Commercial Handheld Equipment			
Average Lifespan (yr)			2.2
Average Usage (hr/yr)			261
Average Horsepower			3
Fuel Saving			
Baseline Fuel Consumption (lbs/bhp-hr)			1.35
Improved Fuel Consumption (lbs/bhp-hr)			0.945
Fuel Density (lbs/gal)			6.25
Fuel Price ($/gal)			1.35
Lifetime Fuel Saving (lbs)			698
Lifetime Fuel Saving (gal)			112
Fuel Saving ($)			151
Emission Reductions			
	Emissions (g/bhp-hr)		
	HC	CO	PM
Average Tier I Engines (1995)	151.25	313.1	4
	50	130	0.25
Emission Reduction (g/bhp-hr)	101.25	183.1	3.75
Emission Reduction (tons/unit)	0.192	0.347	0.007
Incremental Costs			
		Low	High
Incremental RPE		$39	$66
Incremental Consumer Cost (w/ Fuel Saving)		-112	-85
Cost Effectiveness for Incremental RPE ($/ton)			
All Costs Allocated to HC Emissions		203	344
All Costs Allocated to CO Emissions		112	190
All Costs Equally Split Between HC and CO: HC Emissions		102	172
All Costs Equally Split Between HC and CO: CO Emissions		56	95
Cost Effectiveness with Fuel Saving ($/ton)			
All Costs Allocated to HC Emissions		-583	-442
All Costs Allocated to CO Emissions		-322	-244
All Costs Equally Split Between HC and CO: HC Emissions		-291	-221
All Costs Equally Split Between HC and CO: CO Emissions		-161	-122

Table 18: Cost-effectiveness of ARB Tier II emission standards for residential handheld equipment.

Cost Effectiveness Analysis: Residential handheld Equipment			
Average Lifespan (yr)			5.0
Average Usage (hr/yr)			9.5
Average Horsepower			1.5
Fuel Saving			
Baseline Fuel Consumption (lbs/bhp-hr)			1.35
Improved Fuel Consumption (lbs/bhp-hr)			0.945
Fuel Density (lbs/gal)			6.25
Fuel Price ($/gal)			1.35
Lifetime Fuel Saving (lbs)			29
Lifetime Fuel Saving (gal)			5
Fuel Saving ($)			6
Emission Reductions			
	Emissions (g/bhp-hr)		
	HC	CO	PM
Average Tier I Engines (1995)	151.25	313.1	4
	50	130	0.25
Emission Reduction (g/bhp-hr)	101.25	183.1	3.75
Lifetime Emission Reduction (tons/unit)	0.008	0.014	0.0003
Incremental Costs			
		Low	High
Incremental RPE		$39	$66
Incremental Consumer Cost (w/ Fuel Saving)		33	60
Cost Effectiveness for Incremental RPE ($/ton)			
All Costs Allocated to HC Emissions		4915	8317
All Costs Allocated to CO Emissions		2718	4599
All Costs Equally Split Between HC and CO: HC Emissions		2457	4159
All Costs Equally Split Between HC and CO: CO Emissions		1359	2300
Cost Effectiveness with Fuel Saving ($/ton)			
All Costs Allocated to HC Emissions		4129	7531
All Costs Allocated to CO Emissions		2283	4164
All Costs Equally Split Between HC and CO: HC Emissions		2064	3766
All Costs Equally Split Between HC and CO: CO Emissions		1142	2082

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APPENDIX A: SOLICITATION LETTER

October 24, 1995

TO: ALL MANUFACTURERS OF UTILITY AND LAWN AND GARDEN EQUIPMENT ENGINES, AND ALL OTHER INTERESTED PARTIES

SUBJECT: Solicitation of Information on Technical Feasibility and Incremental Cost of the 1999 Emission Standards

Engine, Fuel, and Emissions Engineering, Inc. (EF&EE) has contracted with the California Air Resources Board (ARB) to carry out a detailed literature review to identify emission control technologies for handheld equipment to meet the proposed 1999 emission standards, as well as the costs of meeting those standards.

One of the tasks in the project is to assess the incremental developmental and production costs of low-emission, handheld equipment on a large-volume production basis. We are interested in obtaining your comments on the technical feasibility and incremental manufacturing costs of the following technologies:

a) Four-stroke engine with overhead valves (OHVs);
b) Four-stroke engine with OHVs and exhaust gas recirculation (EGR);
c) Four-stroke engine with OHVs, EGR and catalytic converter;
d) Indirect fuel injection or IDFI (cylinder wall or transfer ports) two-stroke engine;
e) IDFI two-stroke engine with catalytic converter;
f) Direct fuel injection (DFI) two-stroke engine;
g) DFI two-stroke engine with catalytic converter;
h) Other technologies (please specify).

This information will assist us to develop a credible incremental manufacturing cost. We are interested in your views on the technical feasibility of the listed technologies, as well as the effects and impacts of these technologies to issues associated with the development, manufacturing, and performance on handheld equipment in general or on a specific type of hand held equipment.

In addition, we are interested in your comments and suggestions for how we can best translate changes in incremental manufacturing costs into retail price equivalents (RPE). We presently consider that the best approach may be to adapt the RPE methodology used by EPA for automotive technologies. A somewhat simplified explanation of this methodology as applied to automotive technologies is given below. Please comment on whether the basic methodology is applicable to utility equipment, or what changes might be required to make it applicable, and on what the appropriate values for the different parameters might be.

We would like to thank you in advance for your inputs. Please contact me or Mr. Lit-Mian Chan of my staff if you have questions concerning this request.

Sincerely,

Christopher S. Weaver, P.E.
President

Retail Price Equivalent Methodology
(source: Weaver and Turner, 1993)

To bring some order and reproducibility to cost estimates of emission control systems, EPA has developed a standard retail price equivalent (RPE) technique (Jack Faucett Associates, 1985). We applied this technique, along with publicly-available cost estimates by GM and Camet (an EHC supplier), to estimate the incremental cost of an EHC in a typical vehicle. EPA's RPE methods were first outlined by Lindgren (Lindgren ,1978) in a study done for EPA in 1978, and refined by Putnam, Hayes, and Bartlett (PHB, 1984). The present study utilizes an adapted version of PHB's method, with updated information on manufacturer overhead and profit margins developed in a 1985 study by Jack Faucett Associates (Jack Faucett Associates, 1985). The basic equation for the retail price equivalent (RPE) of a given vehicle modification is given by:

$$RPE = ((SP + AL + AO) * MM + RD + TE + WC) * DM$$

where:

RPE is the retail price equivalent;

SP is the supplier price charged to the auto assembler for the components and subassemblies involved;

AL is the direct cost of assembly labor for installing the components;

AS is the manufacturer's assembly overhead cost per unit;

MM is the manufacturer's markup percentage, to account for corporate overhead and profit;

RD is the manufacturer's research and development cost per unit;

TE is the manufacturer's tooling cost per unit

WC is the manufacturer's added warranty cost, per unit; and

DM is the dealer's markup percentage.

The 1985 study by Jack Faucestt Associates estimated the manufacturer's and dealers' markup percentages. For light-duty vehicles, the markup factors recommended by JFA were 19.2% for manufacturer's markup, and 5.7% for dealer markup. For the manufacturer's assembly overhead, PHB estimated roughly 40% of direct labor costs.